AN ASSESSMENT OF THERMAL ENERGY STORAGE AND WASTE HEAT DISSIPATION WITH TOTAL ENERGY SYSTEMS FOR MIT

James Duane Palmer

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by

JAMES DUANE PALMER

B.S., University of Texas at Austin
(1970)

SUBMITTED IN PARTIAL FULFILLMENT
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AN ASSESSMENT OF THERMAL ENERGY STORAGE AND WASTE HEAT DISSIPATION WITH TOTAL ENERGY SYSTEMS FOR MIT

by

JAMES DUANE PALMER

Submitted to the Department of Mechanical Engineering on December 20, 1977, in partial fulfillment of the requirements for the Degrees of Master of Science in Mechanical Engineering and Nuclear Engineering.

ABSTRACT

Total energy systems have been proposed for installation at MIT. Competing power plant configurations based on three different prime movers; steam turbine, gas turbine, and internal combustion engine are analyzed to determine their coincident electrical and thermal power generation capacities. Power generation and demand profiles are compared and methods to match these profiles are formulated. Thermal energy storage is considered as a means of decoupling the thermal power production and demand. The waste heat rejected from each plant configuration is determined. Systems for dissipation of this waste heat are addressed and evaluated to determine their applicability at the MIT site. Configurations incorporating each of the prime movers with an optimal waste heat dissipation system are proposed for detailed simulation of operation and cost comparison.

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I. INTRODUCTION

The consideration of thermal energy storage and waste heat dissipation system options has arisen from a systematic approach to the conceptual design of total energy systems for use in feasibility studies.

This study assesses the potential for application of thermal energy storage in the conceptual design of a total energy system supplying the power demands of the campus at MIT. The economic impact of waste heat dissipation on the alternative conceptual designs is determined.

1.1 Conceptual Design of Total Energy Systems

Total energy systems are on-site electrical generation plants with heat recovery and/or generation supplying all or part of the energy demands of a complex (1). These installations do not incorporate sewage treatment or refuse disposal as would integrated utility systems. Strictly speaking, a better term for the plants being considered is co-generation facilities, meaning installations where both electrical and thermal power are generated in usable forms simultaneously. However, in some circles this term has come to imply facilities which sell or exchange power to or with a distribution network outside the complex served (2, 3). To avoid this implication and to limit the systems studied to those which will be called upon to supply, at a maximum, a given complex's energy demands, the term 'total energy' is used.

The basic incentive to consider on-site generation is the possible margin between the cost of power purchased as electricity and the operating cost of the facility required to generate the same power. When this margin is large enough to defray the capital expenditure involved, the decision to construct on-site facilities is indicated.

With the total energy system concept, this incentive margin is expanded to encompass the difference between the cost of all the energy consumed in a complex as it is conventionally purchased and the cost of the same energy purchased as the fuel for a single plant producing energy in the multiple forms required by the complex. When the possible existence of this margin has been confirmed through preliminary calculations (4, 5), a detailed feasibility study is necessary to accurately assess the size of the margin, the capital expenditure, and thus the possible return on investment or savings with construction of the proposed facilities. To determine and compare the operating costs of possible total energy system configurations it is necessary to establish a conceptual design for each of the alternatives, formulate an operational model for each option, and then perform a simulation of its operation while matched to a load or demand model of the complex to be served.

To reduce the number of total system alternatives to be evaluated by computer simulation some optimization or elimination of alternatives can be accomplished at the conceptual design stage. This is a common procedure in

systems analysis and is usually accomplished by considering the total system to be composed of interrelated subsystems each performing a particular function. Subsystem options for the accomplishment of one function are then identified and grouped according to their compatibility with the different but interrelated subsystems of the total facility. Each group of subsystem options so formed may be evaluated independently of the rest of the system, provided that each subsystem presents the same constraints on the remainder of the total system. With a single criterion for evaluation, such as capital cost, each group can be reduced to only one subsystem which will contribute the minimum initial cost to the complete facility. The facility alternative which is composed of subsystems similarly optimized and compatible with one another will then be the optimum system within the original set of subsystem interrelations. In this manner it is only necessary to perform computer simulation of those alternative systems with dissimilar interrelations among subsystems. With multiple and more complex criteria for evaluation, the grouping of subsystem options in this method will highlight the necessary tradeoffs and their effect on total system operation.

With this systems analysis approach the conceptual design of a total energy system begins with the division of the system into appropriate subsystems. Definition of the appropriate sybsystems can be accomplished in a number of ways. This is a management decision which will be based on

the design group's organization, experience, and available data base. The subdivision should follow functional lines such that the purpose or function of each subdivision can be clearly defined. Regardless of the subdivision of the basic power plant it is necessary to integrate the power plant with the environment, which provides the ultimate heat sink, and to match the power plant's generated power to that demanded by the load. For establishment of total energy facilities at an existing complex the load or power requirements of the complex and the environmental conditions are specified, requiring only analysis and modeling prior to computer simulation. Design options must be considered for the plant/load and plant/environment interfaces as well as the power plant itself. The options at these interfaces are considered in Chapters II and III respectively.

1.2 Conceptual Design Project - A Total Energy System at Massachusetts Institute of Technology

The Massachusetts Institute of Technology, MIT, is considering installing total energy facilities at its campus in Cambridge, Massachusetts. The initial feasibility study is being conducted within the format outlined above. At the research upon which this thesis is based was conducted for the MIT project, the numerical calculations, models, and conclusions presented here relate specifically to the proposed MIT installation; however, care has been taken to include a description of all interface options considered applicable to the general case. The motivation for estab-

lishing a total energy system at MIT is not really unique, nor are the physical constraints presented by the existing complex.

The management of the Physical Plant Department at MIT is faced with the probability of a public utility rate structure change in addition to continued increases in the cost of oil and electricity. Significant effort toward energy conservation has been applied within the existing supply and distribution configuration. The result of this program has been a substantial reduction in the intensity of energy use on campus (6). However, as the "easy" conservation measures have been applied, it is felt that further efforts within the existing configuration will be capital intensive and result in less dramtic decreases. The logical alternative is, then, to modify the existing supply and distribution configuration.

MIT purchases all of its electrical power from the local public utility company. Approximately 80% of campus building heating is supplied by 200 psi steam generated in a relatively modern central facility which also includes steam turbine driven chilled water facilities supplying about 54% of the campus air conditioning load. Of the remaining 20% of the heating load, 15% is carried by individual oil-fired units located in the buildings served, and 5% is purchased as low pressure, 10 psig steam from the local utility. The remainder of the installed air conditioning is in the form of small building or even individual room units distributed about the campus. A detailed analysis of the campus thermal and electrical loads is available in Ref. 7. The constraints presented by the load

with the requirement and methods for matching the plant's output to the load's demand are discussed in Chapter II.

The design information flow in the feasibility study for the MIT project is outlined in the block diagram shown in Fig. 1.1. Since the design load profile information is required for the sizing of the system, this portion of the project has proceeded independently and ahead of the configuration formulations.

One of the major decisions to be made based on the results of the study is what fraction of the electrical load should be supplied from the on-site facilities. Thus, with mode of operation of the plant (peaking, baseload, or total) as a variable in the conceptual designs it is necessary to provide interfacial systems to accommodate a range of plant capacities and demand values. To accomplish this it has been decided to use the latest year of complete demand data, 1976, to form the design basis load and then to formulate systems to provide all or a portion of this load, noting the revisions in the conceptual design necessary to provide for the total demand including growth projections.

The system conceptual design, represented as the "formulate and model alternative system configurations" block in Fig. 1.1, has been broken down into three major subdivisions: the design integration of the power plant with the site and load. The information flow for the conceptual designs is shown in Fig.1.2, where the solid lines represent information on the final models and the dashed lines indicate

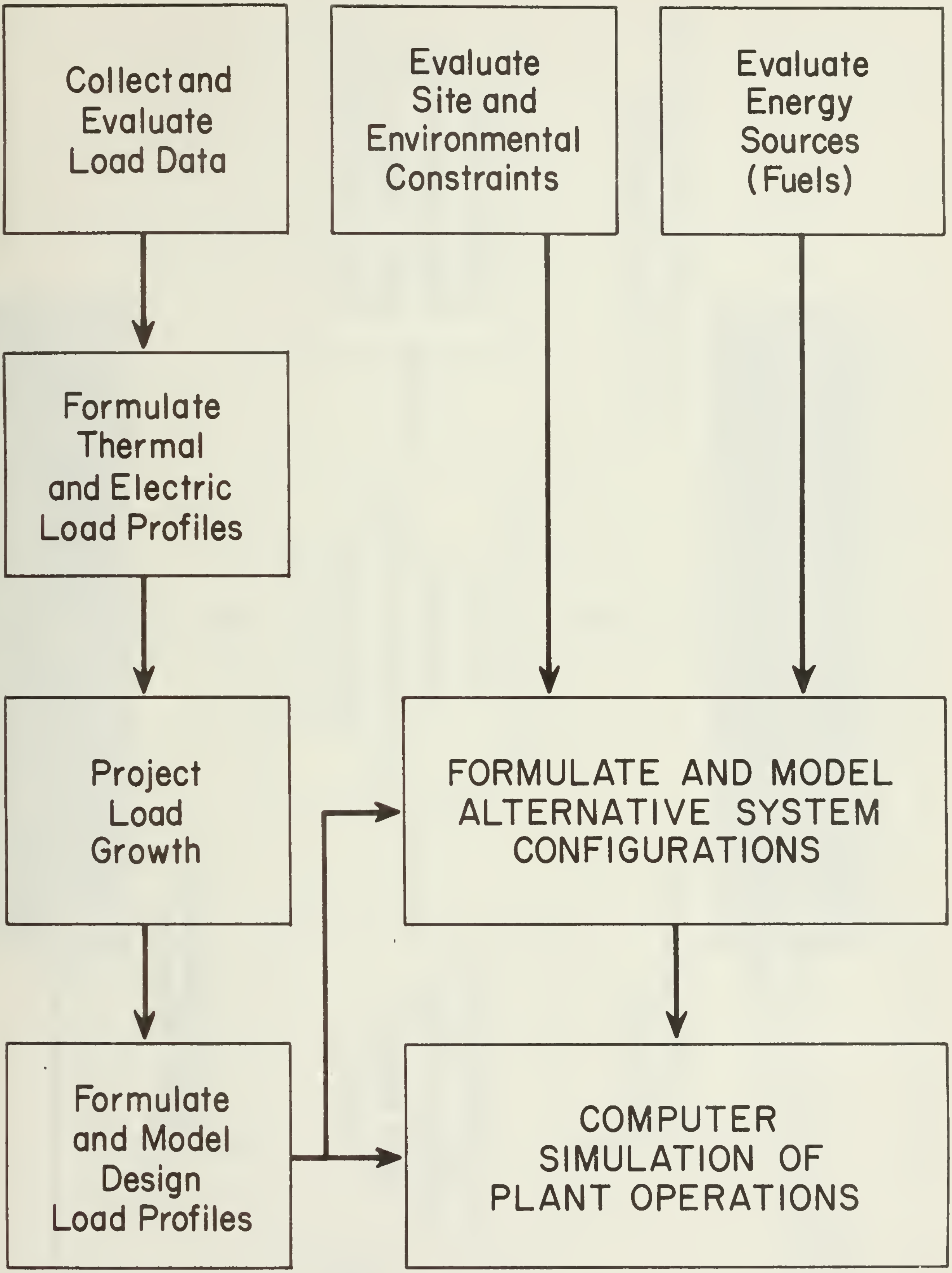

Figure 1.1. Information flow for computer simulation of total energy system operation.

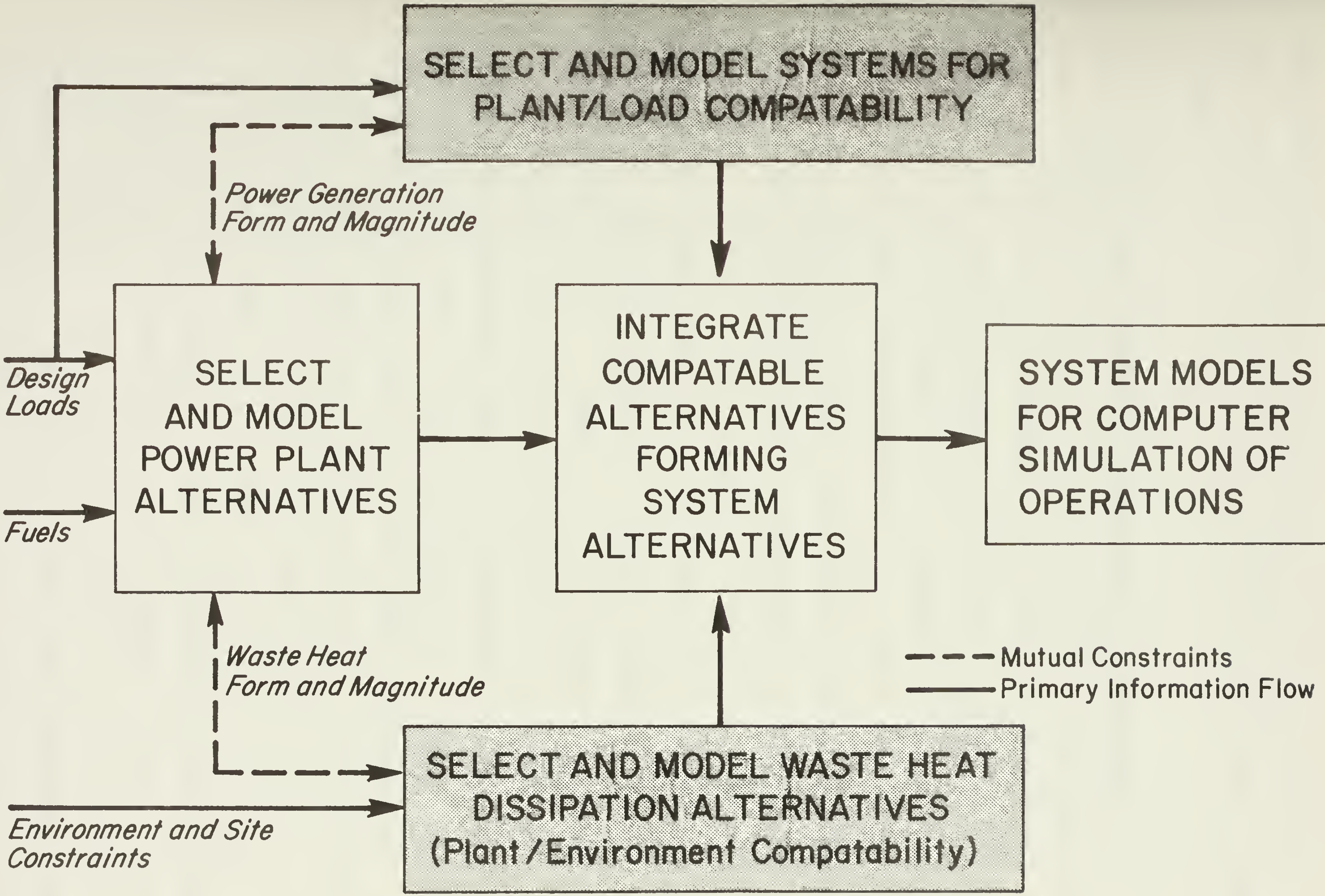

Figure 1.2. Information flow for total energy system conception design and modeling.

exchange of preliminary information on mutual constraints. The portion of the project considered by this thesis is highlighted in Fig. 1.2.

1.3 Method for Definition of Interface Alternatives

Considering the power plant to consist of all that equipment in contact with a working fluid which is necessary to convert the energy of a fuel to electrical and thermal power, the power plant configuration will determine both the magnitude and form of the power generated and the waste heat produced. The load, considered in its basic form of thermal and electrical energy, will determine both the magnitude and form of the power demanded. (The load for a process industry may include a mechanical energy component where there is a need for shaft power to drive plant machinery; however, there is no such requirement for MIT.) When the load is defined in this manner the particular means for conveying and distributing the thermal and electrical power is not specified. The form of the electrical power is best considered as a portion of the generator selection within the power plant configuration options. This leaves the fluid and its state for the thermal power as alternatives at the plant/load interface.

The power plant for the total energy system generates both thermal and electrical power. In efficient power plant configurations the magnitudes of the thermal and the electrical power generation split cannot be expected to match exactly the thermal/electrical load split; therefore the plant/load interface must include systems to match power plant output to the

load's demand in an efficient manner.

Conceptual design at the plant/load interface is, then,a mating or matching procedure with both the input and output constraints or alternatives specified. The plant/environment interface is slightly different since only the magnitude and form of waste or rejected heat is specified. The output consideration at this interface is, then, to minimize adverse environmental impacts while dissipating the waste heat and, of course, to comply with existing regulations and constraints for plant siting.

The results of the load analysis provide the requirements for the output at the plant/load interface. It is necessary to establish preliminary power plant configuration alternatives specifying the generated power split and magnitude for use as an input constraint at the plant/load interface. Options will then be formulated for systems at the plant/load interface which are compatible with each different set of input and output constraints. These alternatives will then be evaluated with reduction to a single alternative for each unique set of constraints where possible or with specific evaluations of the tradeoffs involved where more than one attractive alternative exists. Alternatives will then be modeled for inclusion in total system computer simulation. This study, which results in consideration of thermal energy storage, is presented in Chapter II.

Similarly, at the plant/environment interface the power plant configurations will establish the waste heat rejection

rate and the form, state, or temperature of rejection. Alternative means of dissipating this heat will then be selected for each configuration, with subsequent modeling of the attractive options. Considerations at this interface are contained in Chapter IV.

II. PLANT/LOAD INTERFACE - THERMAL ENERGY STORAGE

Analysis of the requirements for systems which will match a plant's thermal and electrical power output to the demand presented by the load has dictated consideration of thermal energy storage.

2.1 The Plant/Load Interface Constraints

To establish the conditions at the plant/load interface, both the power plant and load must be well defined. These definitions should include;

a. specification of the power generation and demand profiles at various generation rates and times respectively;

b. the media and their states in which the energy is available from the plant and required for the distribution network; and

c. the configurations, with their variable aspects, of both the generating plant and the load.

Some iteration with the power plant design is necessary to arrive at potentially attractive power plant configurations for use as input at the plant/load interface.

2.1.1 The Load or Demand

A knowledge of the load or demand characteristics is necessary to size and evaluate the performance of total energy systems. In completely new installations where the complex being served has not yet been constructed there are methods

available for projecting the power demand (8, 9). Even though the demand data for new installations so obtained are tentative and approximate, the system design may be made more effective since the entire installation can be tailored "from the ground up" to conserve and utilize energy efficiently. Where the system is to be installed at an existing complex, obtaining load data upon which to base plant designs and load models is a relatively straightforward procedure.

For the MIT project hourly load data were available from the records of the Physical Plant Department in the form of total campus electrical load in kilowatts (kw) and total steam flow from the central heating and cooling facility in pounds mass of steam per hour (lbm/hr). It is desirable to characterize both the electrical and thermal loads in consistent energy units. This has the advantage of maintaining the option of considering the thermal load supplied by a fluid other than steam. With steam supplied by the central facility at 200 psig, 420°F and condensate return at an average temperature of 160°F with 8% condensate makeup at 50°F, the enthalpy drop across the thermal load is approximately 1111 BTU/lbm steam, or .3255 KW(t)-hr/lbm steam. Use of this conversion to relate lbm/hr steam flow to thermal load includes the transmission losses in the load value and assumes those losses and the energy and use efficiency to be constant, regardless of the medium of distribution and its temperature. The assumption is, of course, invalid and for the initial conceptual design, the projected error of less than 5% would be considered with

distribution media other than steam; however, further analysis, discussed below, indicates steam to be the desired thermal power distribution medium. For this case the conversion of units represents no error.

A detailed analysis of daily thermal and electrical load profiles has been performed with the 1976 usage data. This analysis and the resulting load model for the computer simulation is contained in Ref. 7. For conceptual design of systems at the plant interfaces, it has been decided to employ six separate days' profiles representing bounds on the hourly thermal and electrical loads.

The extreme conditions or bounds on the generating capacity of the power plant are the maximum and minimum thermal and electrical energy outputs which will be required as inputs to the load. Any total energy system must contain configurations or operating modes to satisfy the demand at these bounds. Additionally, the ratio of thermal power to electrical power produced should be compatible with the same ratio for the power demanded. This requirement dictates the use of maximum and minimum ratios of thermal to electrical power demand as extreme conditions to be satisfied. Determination of these ratios can involve a large amount of data reduction; however, noting that for the MIT complex thermal energy demand is always greater than the electrical demand, the maximum and minimum ratios correspond to the maximum and minimum differences between thermal and electrical power demand. This simpler data manipulation is used to obtain

two additional bound conditions presented by the load.

The extreme conditions employed are determined on the basis of hourly demand (extreme hourly conditions) and then, for energy storage considerations the demand on the day when the extreme occurs is used as a data base. This is not entirely precise since energy storage is a cumulative or integral process over time. There may be a more extreme condition -- that of a larger (or smaller) ratio of the integrated thermal to electrical demand over some time period than that of the day with the maximum (or minimum) hourly difference between thermal and electrical load. The computer simulation of system operation will point out these instances, and the conceptual design can be adjusted as appropriate.

For the latest year of available data, 1976, the conditions for MIT which are selected as extremes for the design, and the dates on which they occurred are:

a. maximum hourly electrical load - 13 August 1976;

b. maximum hourly thermal load - 23 January 1976;

c. minimum hourly electrical load - 22 May 1976;

d. minimum hourly thermal load - 12 September 1976;

e. maximum difference (thermal-electrical) load - 23 January 1976; and

f. minimum difference (thermal-electrical) load - 12 September 1976.

Since the thermal load at MIT is much larger and varies over a greater range than the electrical load, the maximum and minimum differences occur on the same dates as the maximum and minimum thermal demands respectively. Thus, only four

days' data are needed in order to bound the daily profiles. The load data for these dates are included in Appendix A and plotted in Fig. 2.1.

Several general observations on the characteristics of the load are possible from the extreme daily profiles in Fig. 2.1. Some of these are:

1. Maximum daily demands occur during the working day even on the weekends, though on the weekends the demand fluctuation is less. The load at MIT, therefore, is more characteristic of a commercial facility than of a residential complex. This is reasonable since more building area is devoted to instruction, administration, and research than to dormitory facilities.
2. Thermal demand varies over a wider range than electrical demand.
3. The effect of solar heating of the buildings which would be characterized by a predominant afternoon dip in thermal demand is, at most, minor. This is reasonable since most buildings at MIT are of large masonry construction presenting large volume to surface area ratios.

These observations are general in nature and serve only to characterize the type of load being considered.

To describe the load fully, annual profiles were constructed using monthly averages of the hourly thermal and electrical demands. These profiles are presented in Fig. 2.2 plotted from the data tabulated in Appendix A. With the

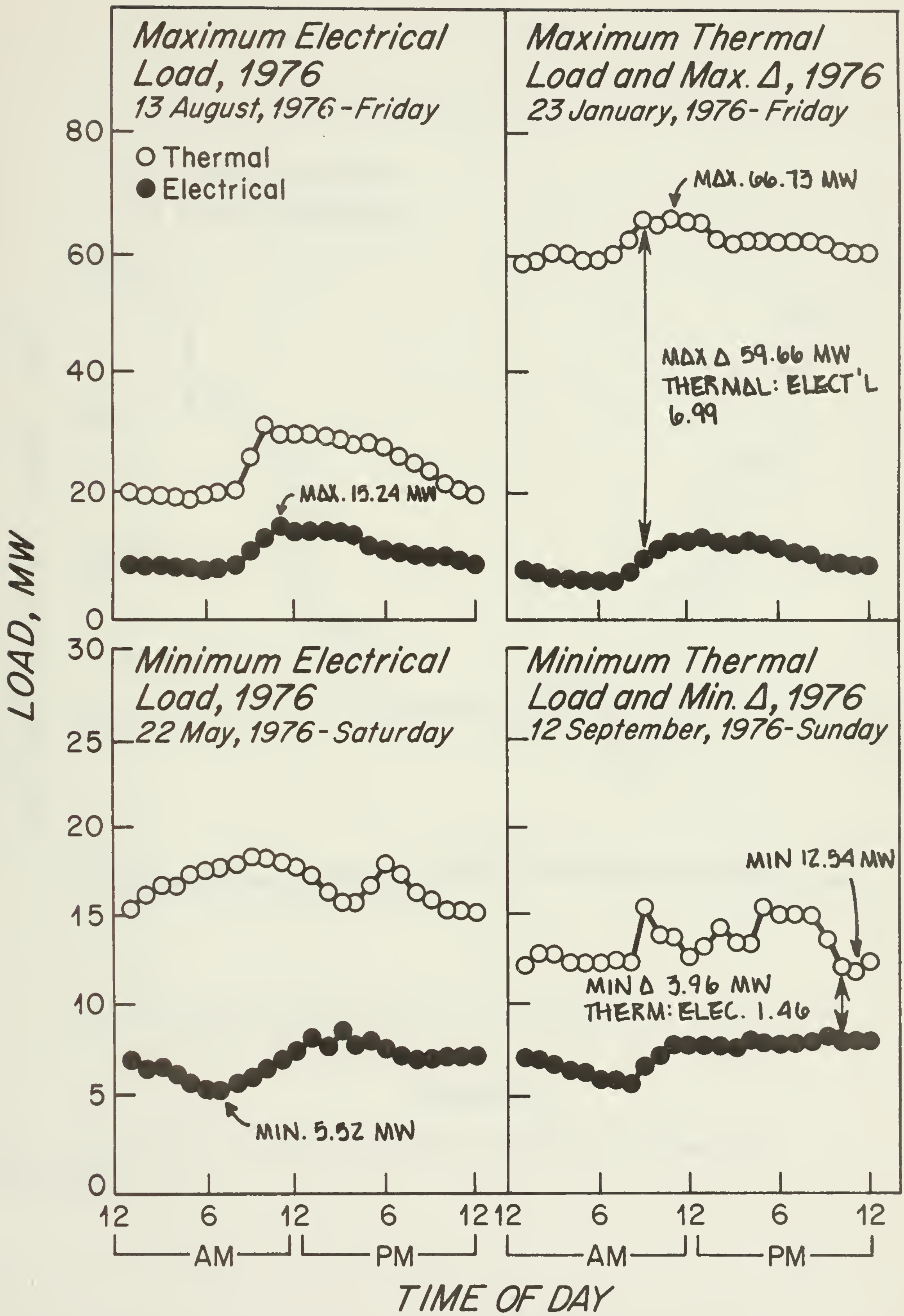

Figure 2.1. MIT daily load profiles for extreme conditions during 1976.

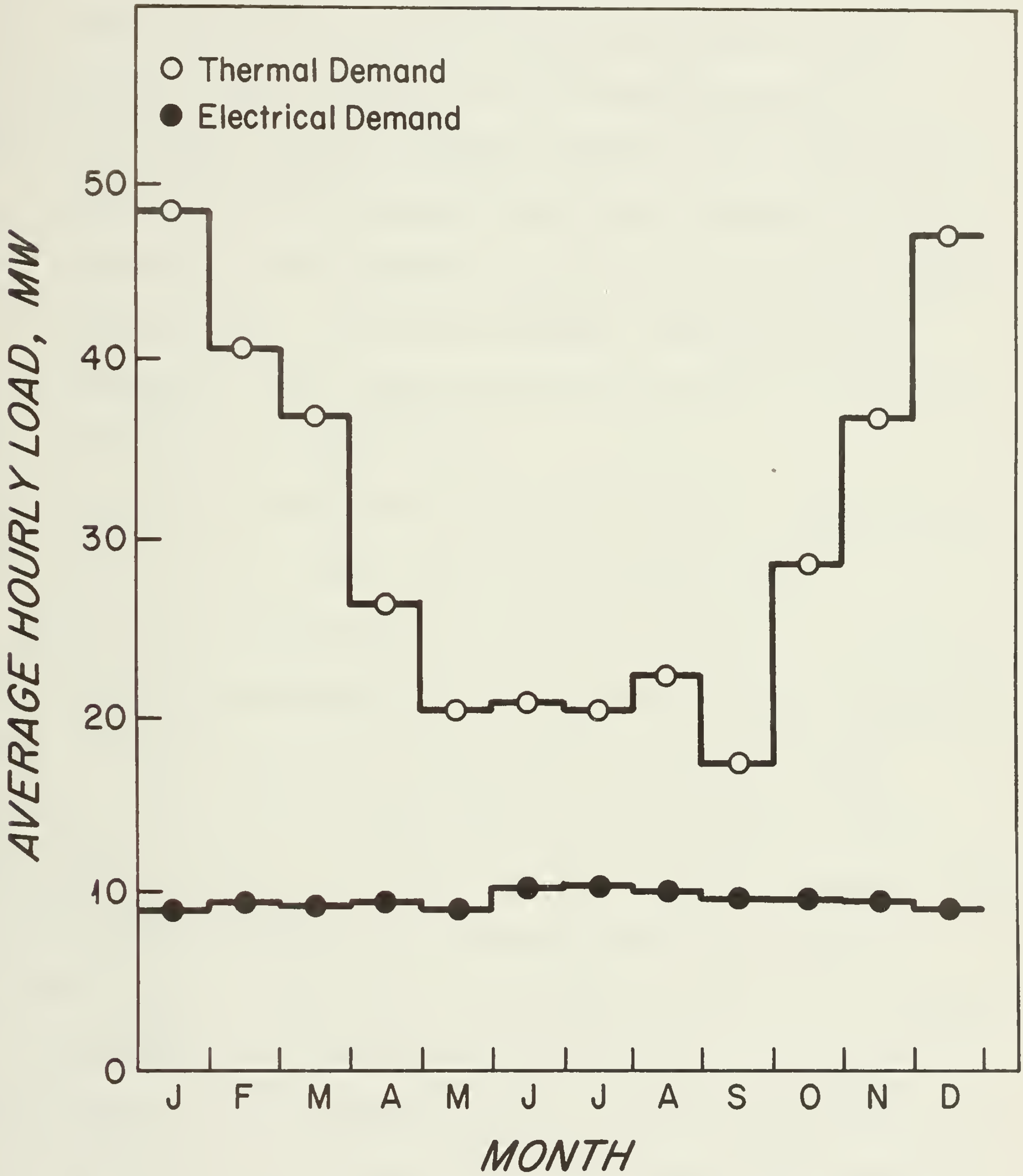

Figure 2.2. MIT annual load profile--monthly averages of thermal and electrical demand.

average meteorological data also included in Appendix A, the following general observations can be made:

1. The thermal load used for space heating is much greater than that used for air conditioning.
2. The larger average thermal demands occur coincident with the smaller average electrical demands. This is opposite to the desired variation for a total energy system since, for these systems, less electrical generation will characteristically result in smaller thermal power production.
3. As with the daily profiles, the thermal demand varies over a wider range than the electrical load.

The 1976 load profiles can provide a basis for comparison of the operating performance of the various system designs; however, a more realistic evaluation is possible using projected load profiles indicative of the expected power demand over the lifetime of the system. Accurate energy demand projections are difficult to formulate and are worthy of detailed study in themselves but, regardless of the estimates, once a system has been designed and installed the growth in energy usage by the expanding complex can be controlled or at least influenced by the capacity of the installed generation system.

Energy demand growth for the MIT complex was projected based on planned building construction until 1985, estimated construction for 1985 to 2000, and previous energy use per building type (7). A major unknown factor in the growth

projections is the load reduction in the existing buildings from the application of the Facilities Management portion of the MIT Energy Conservation Program , ENCON. Facilities Management at MIT is performed by a central control and monitoring system consisting of an IBM System/7 Computer with remote control and sensing devices. This system originally installed for programmed control of the operation of space comfort systems in seven major building on campus in 1974, will be expanded to include the majority of major energy consuming buildings on campus prior to 1980. It has been optimistically estimated that this control system expansion plus the refinement of the control programming with operating experience would result in a load reduction of 12% in both electrical and thermal loads relative to the 1976 demand (11).

To illustrate the effect of this conceivable initial load reduction due to Facilities Management, the demand projections summarized in Table 2.1 and detailed in Appendix B include the projected load with and without the 12% reduction of the existing 1976 load. The growth values were projected on the basis of an average load with existing peak to average yearly ratio used to determine the peak load with growth. This results in peak and average loads expanding in equal ratios. With all of the approximations used to account for the many variables, the accuracy of the energy use projections is really unknown; in fact, any reasonable projection utilized in the selection of a system configuration can ultimately constrain the actual energy demand from expansion of the complex.

TABLE 2.1

MIT ENERGY DEMAND PROJECTION TO YEAR 2000

Percentage increase:

	Without Facilities Management		With Facilities Management	
Year	Thermal	Electrical	Thermal	Electrical
1976	--	--	-12.00	-12.00
1980	2.23	7.64	-10.04	-5.28
1985	9.51	21.54	-3.63	6.96
1990	13.71	29.09	0.07	13.60
1995	21.06	43.17	6.53	25.99
2000	23.27	46.29	8.47	28.74

2.1.2 Thermal Energy Distribution Options

With the load defined as an energy demand, it is necessary to specify the medium for distribution of the required energy to completely establish the constraints on the load side of the plant/load interface. The electrical portion of the load is assumed to require the conventional 3 phase 60 hertz, 440 volt alternating current transformed to single phase, 60 hertz at 220 and 110 volts for ultimate use. This form of electrical distribution would be required for most applications due to its universal acceptance in the United States and its use as the assumed power source for the construction of all standard electrical appliances. There has been some success with deviations from this standard specifications especially in the area of increased efficiency with high frequency lighting (11). Incorporation of this innovation requires additional electrical generators and wiring with more complex switch gear. It is felt that for the first experience with on-site electrical generation the electrical systems should be maintained as simple as possible. The feasibility of additional and more elaborate electrical distribution schemes should be investigated concurrent with planning for the expansion of the complex served.

These same arguments, the existence of a proven distribution system with experience in the operation of that system, hold for the thermal energy distribution system employing steam as the medium of distribution. However, much

of the success to date with total energy systems has been in Europe where these type plants are coupled to district heating networks employing high temperature water as the distribution medium (12). Since water distribution systems do not require more complex configurations and their operation is not so different from steam systems as to require extensive retraining of plant operating personnel, conversion to a water system was not considered to be too big a step to make simultaneously with the establishment of a total energy system. Thus, a designer's first impulse -- to retain the existing thermal energy distribution system -- is questioned. For the case of MIT this impulsive decision is proven correct by subsequent analysis but in addition the investigation points out the conditions necessary for advantageous application of steam distribution and for the possible future application of a water distribution system.

Historically, the development of thermal energy distribution systems has followed two distinctly different paths identified consistently by the geographic location of the facility. In the United States steam has been used as the medium for distribution while in Europe water has been so employed. This divergence of technology was due to the differing characteristics of two factors:

a. natural resources -- fuel and water; and
b. load characteristics -- density of energy demand and final form in which energy will be employed.

In Europe both fuel and water have not been as plentiful, and hence as inexpensive, as in the United States. This encouraged European development of more efficient distribution systems requiring higher capital costs, since the fuel savings would more rapidly recoup the additional initial investment costs. In Europe with its generally higher population density there were also larger areas available for the establishment of distribution systems from a central plant with a large portion of the thermal load used for space comfort. The United States has had smaller areas with energy demand densities great enough to warrant the generation of thermal power in a central facility, and most of these areas are industrial with equipment and processes designed to utilize steam. These conditions, especially the availability and cost of fuel and water, have changed, placing the United States' circumstances more in line with the European experience.

As late as 1951 the National District Heating Association stated the following advantages for the use of steam as opposed to water for district heating in the United States (13):

> "If tall buildings are to be served, extremely high pressures must be carried in the distribution system or the consumer must provide pumping facilities.
>
> "Hot water requires a two-pipe street distribution system.
>
> "Hot water distribution systems are less convenient

to repair and more consumers are affected when a shutdown of a portion of the system is required. To repair leaks or connect new customers to the street mains the affected lines must be drained and service to all consumers in the affected area discontinued. It is possible to bypass the affected area if provision has been made with adequate cross connections and sufficient valves. This additional equipment increases the cost of a system which is already high in investment.

> "Capital investment for a hot-water distribution system is generally higher than for an equivalent steam system because of the dual piping required for supply and return mains. The investment should not be assumed to be double, however, since steam traps and pressure reducing valves are eliminated."

This comparison was based on a single-pipe steam distribution system where the hot condensate which resulted from the cooling or release of thermal energy from the steam was simply dumped and not returned to the central plant. Single-pipe operation is only feasible with an abundant water supply and sufficient sewage or dumping facilities. The single-pipe system is also inherently inefficient since the thermal energy of the hot condensate is wasted. The difference between the enthalpy of the dumped condensate and the water supply represents energy supplied by the plant but not utilized. Modern steam distribution systems, including that at MIT, utilize two-pipe configurations whereby the

condensate is returned to the central plant where it is employed as feedwater for the boiler with additional makeup water required only as necessary to balance leakage and blowdown losses.

When a two-pipe steam distribution system is compared to the equivalent high temperature water system, Diskant (14) has shown a 6% savings in capital costs and a 17% reduction in operating expenses for the high temperature water systems. These savings were for a 30 MW peak thermal load with normal daily and annual profiles (similar to those at MIT), supplied by oil-fired high temperature water generators (corresponding to the boilers in a steam installation) producing water at 400°F and 290 psi. The economic advantage in capital cost with high temperature water systems comes from the use of smaller diameter piping since water has a larger volumetric specific heat, and from the absence of steam traps and reducing valves in the high temperature water designs.

The lower operating costs with water systems are due to the decreased losses in the supply piping. Piping losses with a steam system are larger since there is a somewhat increased heat transfer coefficient at the pipe/fluid surface with condensing steam inside the pipe, but more so since the steam distribution piping must include traps to remove the condensate. With the use of steam traps, losses characterized as flash losses occur. These losses are due

to the entrance of steam into the condensate system from the "flashing" of a portion of the hot condensate when it is admitted, by the trap, to the lower pressure return piping. The low pressure steam in the return piping is then either vented off or condensed to ensure proper operation of the liquid condensate pumps. An additional thermal loss in the steam system, termed "blowdown loss", is due to the necessity of removing non-volatile impurities from the boiler. The vaporization process in the boiler allows accumulation of the corrosion products from the feedwater as well as impurities from the makeup. If they are not effectively removed these impurities foul heat transfer surfaces. Removal is accomplished with boiler blowdown where a portion of hot boiler water is dumped with the impurities suspended in this blowdown water. Without a vaporization process in the hot water system there is no continual concentration of impurities and no required blowdown. Thus, it is possible for high temperature thermal distribution systems to be economically advantageous in both aspects -- initial and operating costs.

For the cases where the use of steam is required or has definite advantages in a particular building, European designs incorporate secondary heat exchangers in that building. These heat exchangers allow transfer of thermal energy from the primary high temperature, high pressure water to produce steam on the secondary side. This secondary heat exchanger arrangement isolates the individual consumers from the main supply network. These configurations

allow tall buildings and industrial units to be supplied with steam from the high temperature water distribution main. Some district heating configurations employ the same arrangement with lower temperature water on the secondary side of the heat exchanger. Decoupling the individual buildings from the distribution main facilitates repair to those portions of the system (15). The European experience has generally resulted in numerous advances in the technology for design, construction repair and operation of district heating systems employing high temperature water (16).

High temperature water is an alternative to steam as the medium in a thermal energy distribution network and can be a definite advantage in new installations, but for retrofit applications, a conversion is rarely possible because of the large capital expenditure which is required. Accurate assessment of the cost of conversion is difficult without a detailed design of the proposed system. Information from one of the primary Architect-Engineering firms which has been involved in the design of conversions from steam to hot water distribution systems, (17), leads to the conclusion that these conversions are economically attractive primarily when coupled with a proportionately large expansion of the distribution network. Even the European experience has been to expand hot water district heating networks only to those building which had previously used hot water for heating; it has not proven financially feasible to incorporate the buildings which employed steam

heat (18).

Even without a detailed design and cost analysis for conversion of MIT's steam distribution system to high temperature water it is felt that this conversion alternative must be excluded based on the following factors:

1. Lack of indication that any similar conversion has been attempted without a large scale extension of the distribution network coupled with the fact that no such extension is planned for the MIT system.
2. Consideration of the disruption and its duration which would be required for the conversion. Conversion to hot water heating would require extensive modification of radiators and piping within the entire institute. This conversion would have to be accomplished during the very short, approximately 4 month, period when no building heating is required. The scope and time requirements of the project indicate massive construction disruption over the entire campus.
3. Continued need for steam. MIT chilled water facilities which operate to some extent all year require steam for their operation. Since the chilled water units are relatively new their replacement would be a large economic liability charged to the conversion or, for the case where they would be retained, a plant design would be necessary to produce both steam and high temperature water,

negating some of the cost advantage of a hot water distribution system.

4. Extensive piping modifications internal to buildings. The MIT campus complex has been constructed incrementally from 1916 to the present. Each building has been designed and constructed to the applicable codes and regulations at the time of its construction. It is doubtful that the older buildings' heating systems could be adapted to the use of high temperature water without elaborate refurbishing.

The conceptual design of total energy systems for MIT should, then, consider the thermal load to be supplied in its existing form of 200 psig steam. High temperature water distribution should be retained as an alternative only for the case of expansion of the present campus area requiring extension of the thermal energy distribution network.

2.1.3 Power Plant Configurations

The power plants for total energy systems are usually classified according to the prime mover employed for the electrical generator. They are:

a. steam turbines,

b. gas turbines, and

c. reciprocating internal combustion engines.

The form and method of obtaining thermal power with each of these prime movers is slightly different and can be varied according to the end use application (19). Steam turbines

can be employed as back pressure units yielding low pressure steam or as multi-stage condensing units with interstage steam extraction producing higher pressure steam. The source of thermal power in gas turbine applications is the hot, approximately 900-1000°F, exhaust gases. Utilization of this thermal power may be direct (hot air drying, blowing, etc.) or through heat exchangers producing hot water or steam. Ingenious combined cycles have been developed and employed in larger installations (20, 21). Total energy systems based on reciprocating internal combustion engines enjoy widespread use in the United States, accounting for about 95% of all installations (22). These installations, primarily employing diesel engines, utilize the high temperature (approximately 900°F) exhaust gases in a waste heat recovery boiler for thermal power generation. Additionally, the lower temperature jacket, lube oil, and air cooling functions may be incorporated in feedwater heating schemes or other low temperature applications when the economics of the additional heat exchangers vs. the fuel savings are attractive. The temperatures involved are about 200°F for the jacket cooling, approximately 150°F for the lube oil cooling, and no more than 110°F for air cooling. Again, sophisticated combined cycles have incorporated diesel engine prime movers (23).

The number of general power plant configurations to be considered for the MIT installation has been reduced to three. The preliminary analysis and considerations which

led to the selection of these configurations are presented in Ref. 7. Fig. 2.3 depicts the conceptual configuration for each of the prime movers being considered. These simplified diagrams are not complete plant schematics; only those details sufficient to illustrate power output and rejected heat are included. These plants are represented as providing thermal power in the form of steam. With the exception of the steam extraction plant this is not a necessary distinction since the waste heat recovery devices could also be utilized to provide high or low temperature water to thermal loads. However, as is shown, the plants are the result of one of the first design decisions at the plant/load interface: to retain the existing steam distribution for thermal loads. Implicit in each plant configuration is the option to retain the existing boilers and commercial electrical supply to provide portions of the thermal or electrical power load. These facilities will be used when the configuration is sized to be employed as either a peaking or base load unit.

2.2 Matching of Plant Alternatives to the MIT Load

2.2.1 Extraction Steam Turbine Plant

Operation of the extraction steam turbine offers the capability of varying the thermal to electrical output power ratio within a set range. This is due to the design of the turbine, which will accommodate different flows through the high and low pressure sections to produce the

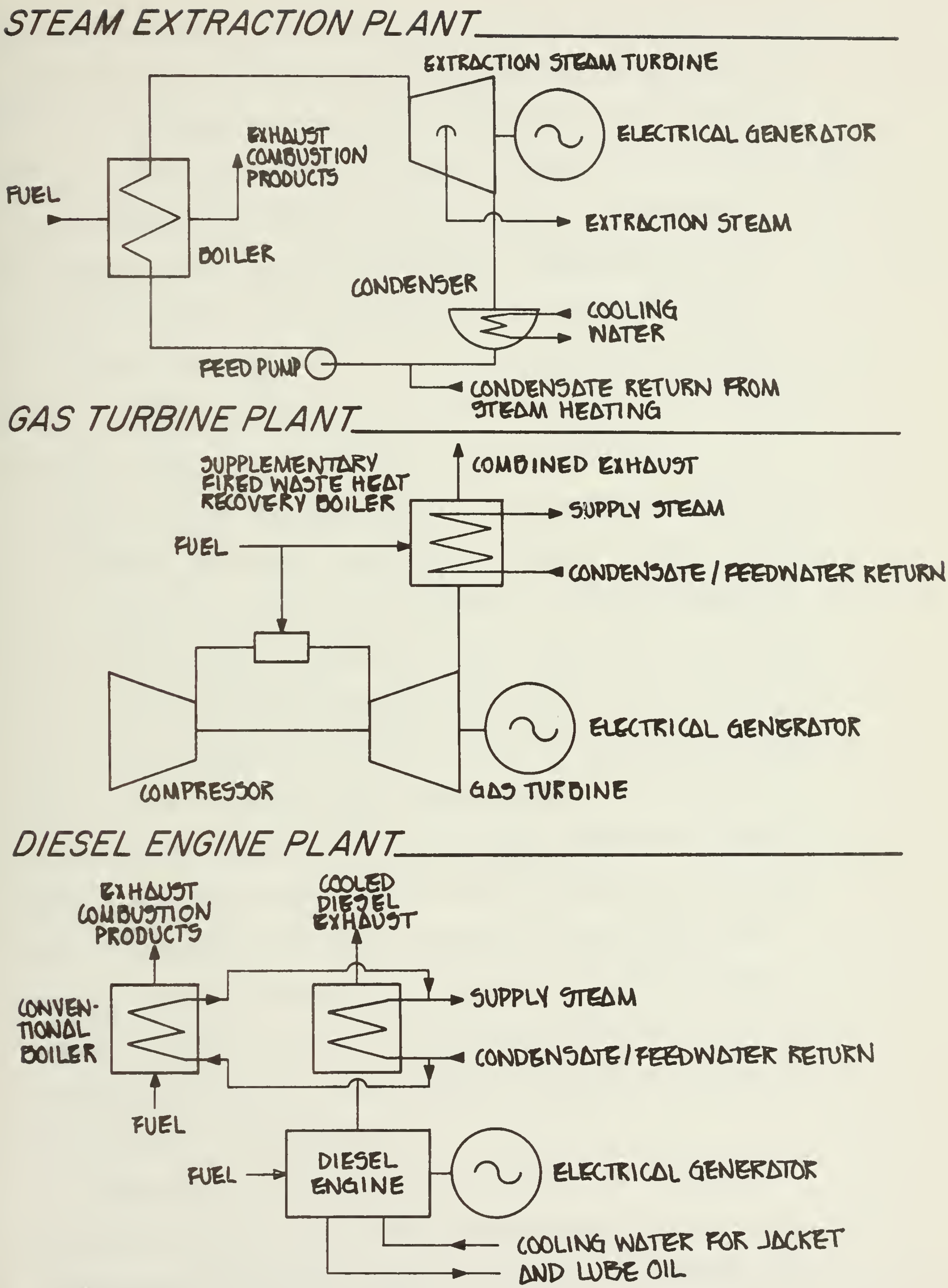

Figure 2.3. Power plant configurations considered for the MIT total energy system.

same shaft power. The flow variation is controlled by the turbine inlet throttle and the extraction steam throttle. Thus, an increase in extraction steam flow will decrease the flow through the low pressure section of the turbine with a corresponding decrease in shaft power; a subsequent increase in inlet throttle flow will increase the flow through both the high and low pressure sections (with extraction flow maintained constant), increasing shaft power. Simultaneously increasing extraction flow and throttle inlet flow in a given ratio can result in an increase in high pressure section flow and a decrease in low pressure section flow with no change in shaft power. These characteristics are exhibited in the performance curves for a 10 MW electrical output rated unit shown in Fig. 2.4. The maximum and minimum flows are determined from boundary layer and turbine blade flow considerations and from the blade cooling requirements, respectively.

With the postulation of a control system to sense thermal and electrical demand and to control the turbine inlet and extraction steam throttle valves within the allowed ranges, the operation of the extraction steam turbine may be simulated. Calculations for the conversion of extraction steam flow to thermal supply in energy units are included in Appendix C.

Comparing the load curves in Fig. 2.1 with the extraction steam turbine performance curve in Fig. 2.4, it is immediately apparent that for electrical demands less

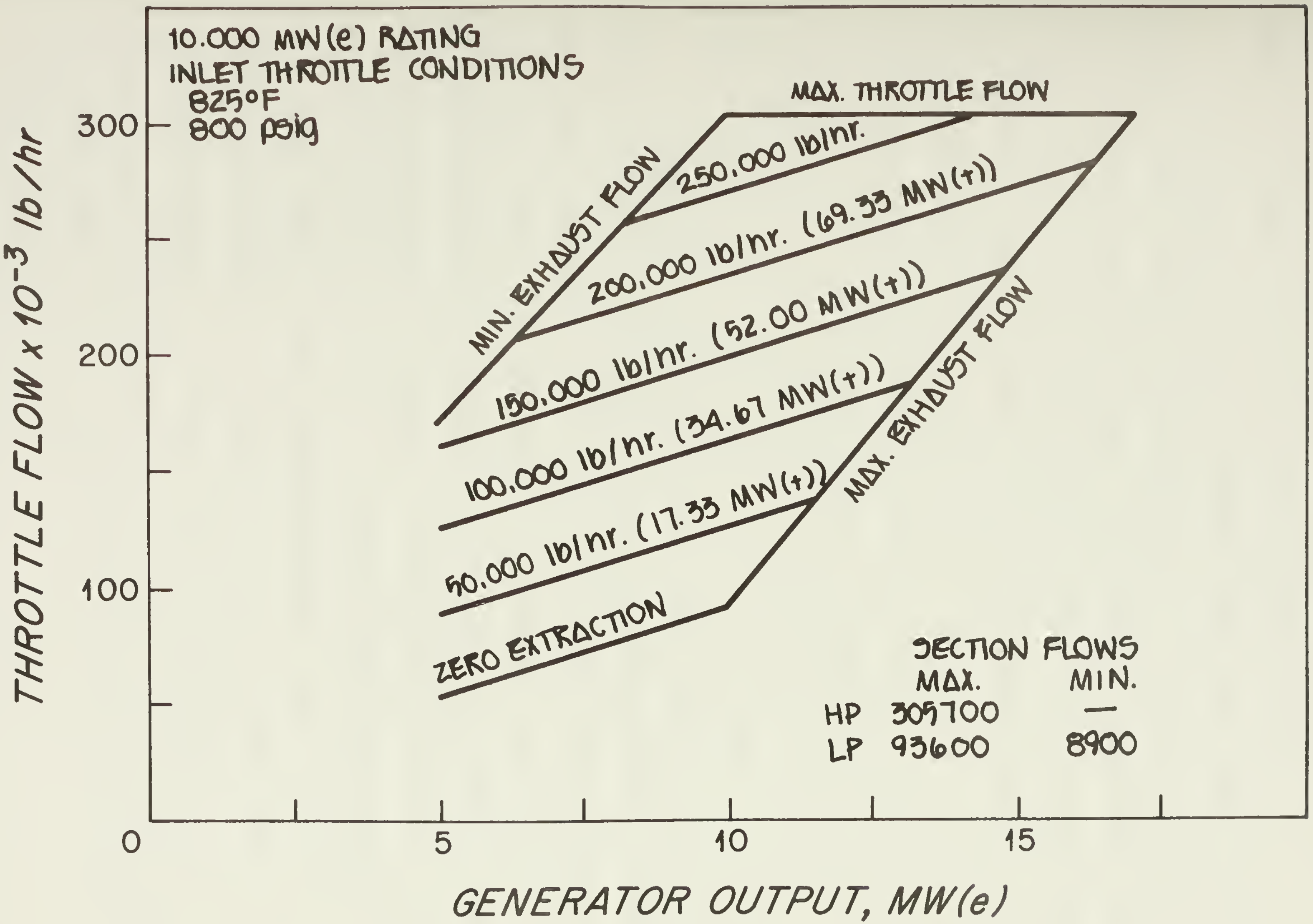

Figure 2.4. Performance characteristics for a 10 MW(e) steam turbine converter with extraction steam at 200 psig.

than 10 MW all thermal loads less than 52 MW can be supplied. Thus, the two minimum load days present no difficulty in regard to balancing power generation and demand.

At electrical outputs greater than 10 MW, Fig. 2.4 indicates that thermal power generation is limited by minimum and maximum extraction flows. The maximum extraction flow allowed at electrical loads greater than about 6 MW exceeds the maximum thermal demand of 66.73 MW (approximately 192,500 lbm/hr extraction flow) and therefore should not be limiting. The minimum extraction flow allosed is seen to vary sharply from 0 at 10 MW electrical demand to approximately 76.27 MW(t) (220,000 lbm/hr) at about 17 MW. This may be limiting - requiring extraction flow greater than the thermal power demand.

To assess this situation an hour-by-hour comparison is made between the load profiles for the days on which the maximum thermal and maximum electrical demands occur and the extraction steam turbine performance on these days. When a limiting condition is present, two different control schemes are postulated to establish the bounds of the mismatch. First, control of the system is assumed to be such that the electrical demand is satisfied, that is, the plant is assumed to operate supplying 100% of the electrical load with excess extraction steam. In the second case, the controlling demand is assumed to be the thermal load. The plant is operated to provide 100% of the thermal demand supplying the maximum portion of the electrical load possible. The deficiency of the electrical power generated compared to the demand is then evident.

The hour-by-hour comparison points out that only on the

day of the maximum electrical load is a limiting situation met, considering just the extreme (boundary) days . This indicates that in certain load situations characterized by electrical loads above 10 MW with low thermal demands, the 10 MW extraction steam turbine plant cannot, by itself, satisfy the load. This situation is illustrated in Fig. 2.5 with control by electrical demand and in Fig. 2.6 with control by thermal demand. Both figures are constructed from tabular data in Appendix C.

These figures show that for the 10 MW extraction steam turbine installation some provision is necessary either to dissipate the excess thermal energy produced which is present with electrical demand employed as the dominant control or to provide the electrical supply vs. demand deficiency which exists with thermal demand used as the overriding control parameter. Neither of these functions could be performed by the existing boiler facilities since a thermal power deficit does not exist. The options which are available to accommodate the mismatch and their ramifications are:

1. Installation of steam dumping facilities with the extraction steam turbine plant. This is inherently wasteful and uneconomic with both a capital cost increase for the additional dissipation facilities and higher operating costs to produce the unwanted energy.

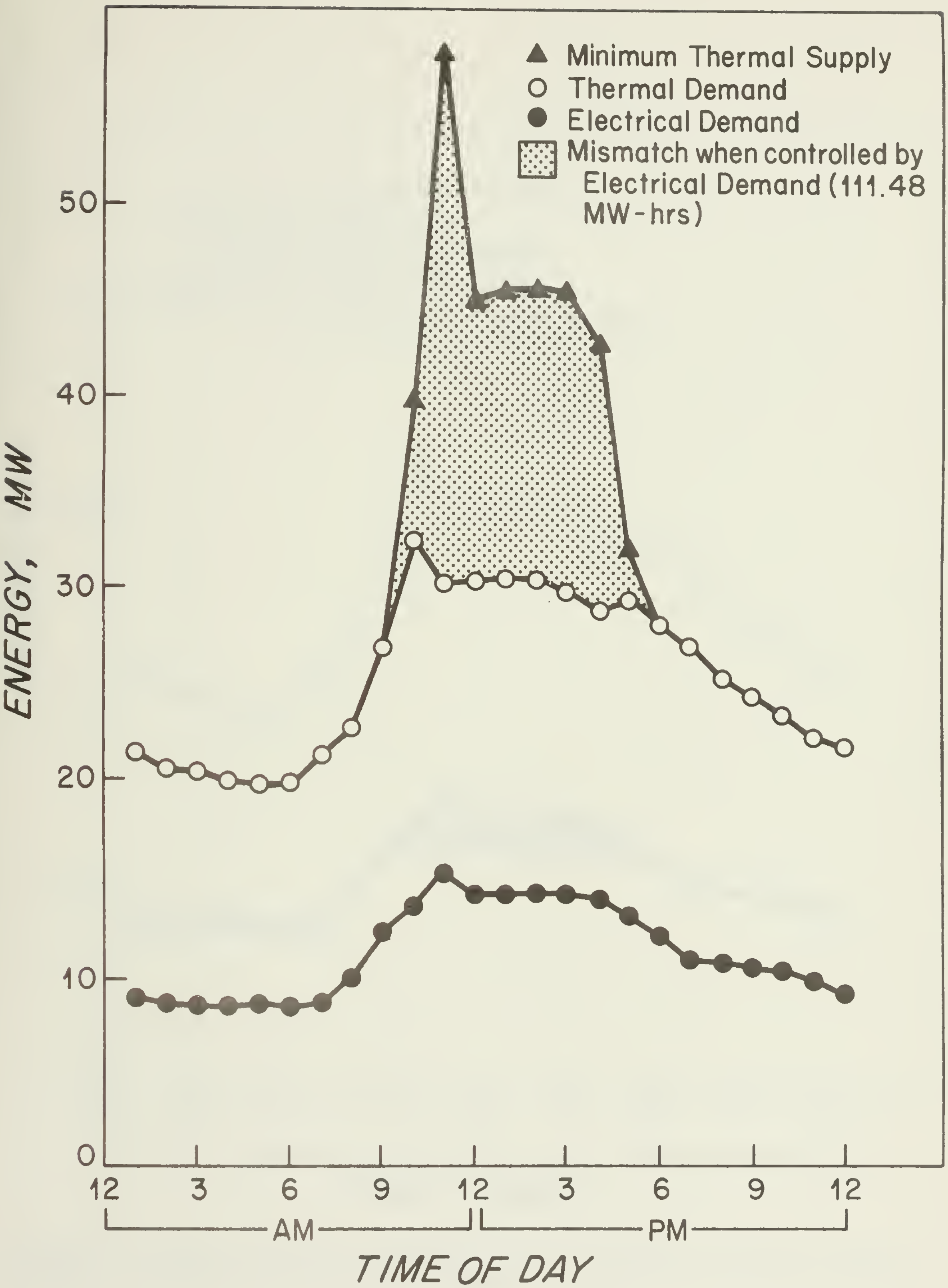

Figure 2.5. Demand-Supply performance for a 10 MW(e) extraction steam turbine plant while controlled by electrical demand.

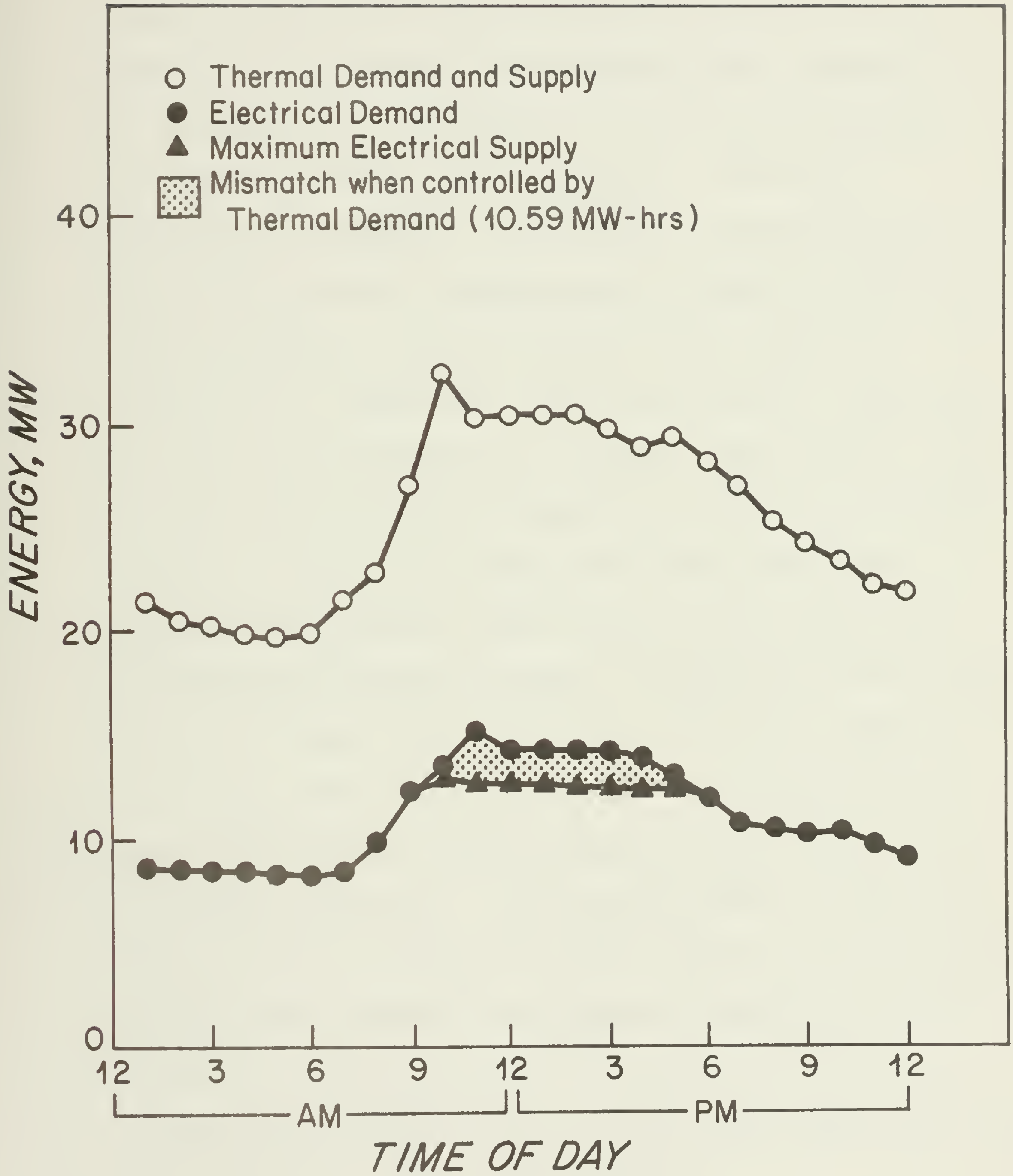

Figure 2.6. Demand-Supply performance for a 10 MW(e) extraction steam turbine plant while controlled by thermal demand.

2. Purchase of electrical power from the utility during periods of electrical power production deficiency. This requires that a tie be maintained with the commercial electrical distribution grid. There will be a charge for maintenance of this tie in addition to the cost of the power purchased. This tie, however, could supply backup power, increasing plant reliability and/or reducing capital cost by removing the necessity to install excess plant capacity.
3. Resizing the extraction steam installation, employing multiple smaller units or a single larger unit. With multiple smaller units operating in parallel to exactly match thermal and electrical power demand the installation would be more complex and require more sophisticated control systems, increasing capital cost. Use of a larger unit which could match the demand at all times would have the unit normally operating at much less that its rated capacity with a decrease in operating efficiency. The excess capacity installed with a single larger unit would result in an increased capital cost without an increase in system reliability since the excess capcity is incorporated in a single prime mover with approximately the same probability of failure as the single smaller unit.
4. Resizing the steam extraction installation employing dual 10 MW units with no tie to the commercial electrical power distribution grid. During periods

when supply from one unit is limited to less than the demand the two units would operate in parallel at less than their rated capacity to exactly match the demand. During periods when demand can be satisfied by one unit the other unit will provide backup increasing system reliability.

The options in 2 and 4 are the most attractive and those recommended by initial study when the steam extraction unit is simulated supplying the 1976 demand.

One additional option for handling the mismatch situation is to store the excess thermal energy developed while matching generated electrical power to the demand. As this thermal energy storage capacity can have other applications and substantial effects on plant design it is considered as a separate topic, 2.3 below.

Revising the magnitude of the load to account for the growth of the complex served over the lifetime of the total energy installation will require either an increase in the rated capacity of the steam extraction turbine or the use of multiple units. Performance curves for large size units differ from those of the 10 MW unit increasing very nearly proportionately in all quantities. This is illustrated with the performance curve for a 15 MW unit shown in Fig. 2.7 where 100,000 lbm/hr extraction steam flow is the minimum for 18.25 MW electrical power generation. Comparing this to 67,000 lbm/hr minimum extraction steam at about 12.17 MW for the rated 10 MW unit the ratio of 1.5 for the rated

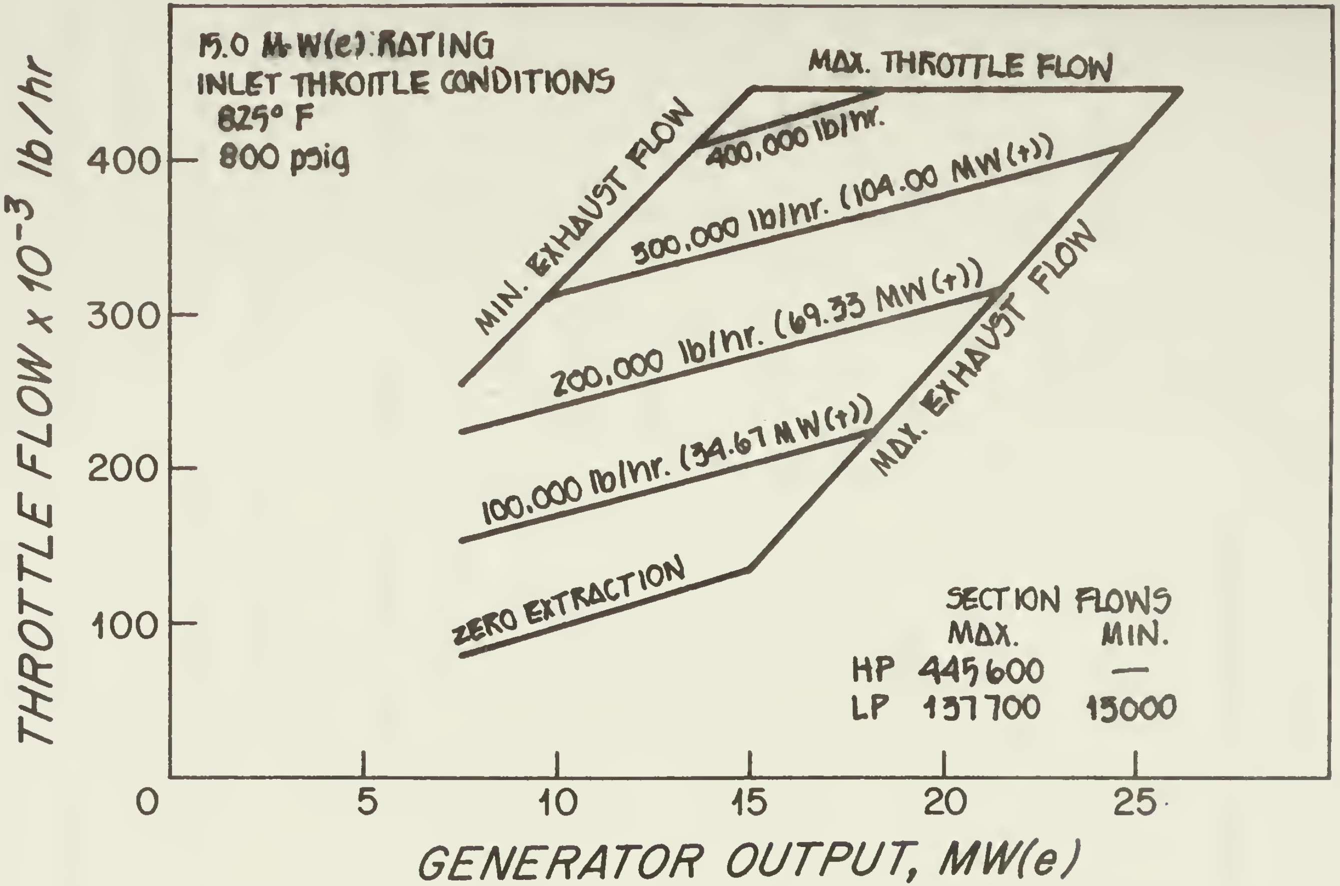

Figure 2.7. Performance characteristics for a 15 MW(e) steam turbine generator with extraction steam at 200 psig.

capacity holds also for minimum extractions flows at the proportional electrical generation. Thus, if the load growth projections were to indicate indentical proportional growth of the thermal and electrical demands, the same limiting situation would be encountered sometime in the life of the system with the installation of a unit whose rated capacity is greater than 10 MW by about the same ratio as the growth projection compared to the 1976 load. Any additional equipment installed to accommodate the excess thermal energy produced (or the deficit of electrical power) would not be utilized over the full life of the system. With the resulting low utilization factor the capital cost of that equipment if installed initially would be an economic liability to the system over most of its life unless it had an alternate function such as providing backup or emergency power as in the case of the tie to the commercial grid. The other alternative for overcoming this economic liability is to install the additional equipment just prior to the time when it would be utilized. This requires a power plant designed to accommodate future addition.

It is doubtful that the electrical and thermal demands will grow unconstrained in identical proportions at MIT since the present campus load represents incremental construction of buildings, laboratories, classrooms, and dormitories dating from 1917 to the present (24). With the technological evolution of construction techniques and available equipment the energy requirements have changed and

will continue to evolve as the complex expands. The unknown factors in this evolution make accurate, long range growth predictions difficult. Additionally, the growth of the energy demand can be constrained, not necessarily in magnitude, but at least in form--electrical vs. thermal-- to that which can be accommodated most efficiently by the energy supply system. This could be accomplished with the prominent input from the Department of Physical Plant into the Institute's expansion and construction planning. Neglecting this effect and applying the percentage growth rates from Table 2.1 with and without the 12% reduction (which is the projected effect of the ENCON Program's Facilities Management) to the limiting 1976 demand condition of maximum electrical load, which occurred at 11:00 on 13 August 1976, a plot is obtained of a possible limiting operating condition as a function of time with projected load increase. These conditons are plotted with the limiting portion of the 15 MW(e) extration steam turbine unit's performance curve in Fig. 2.8. From this figure it is seen that within the validity of the growth assumptions and their application to any one hour's demand, the 15 MW extraction steam turbine will become limited prior to the year 2000 just as the 10 MW(e) unit was limited supplying the 1976 load. The effect of the 12% load reduction attributed to the Facilities Management installation and applied equally to both electrical and thermal demand is to delay the time when the limiting condition is met from the 1980 to

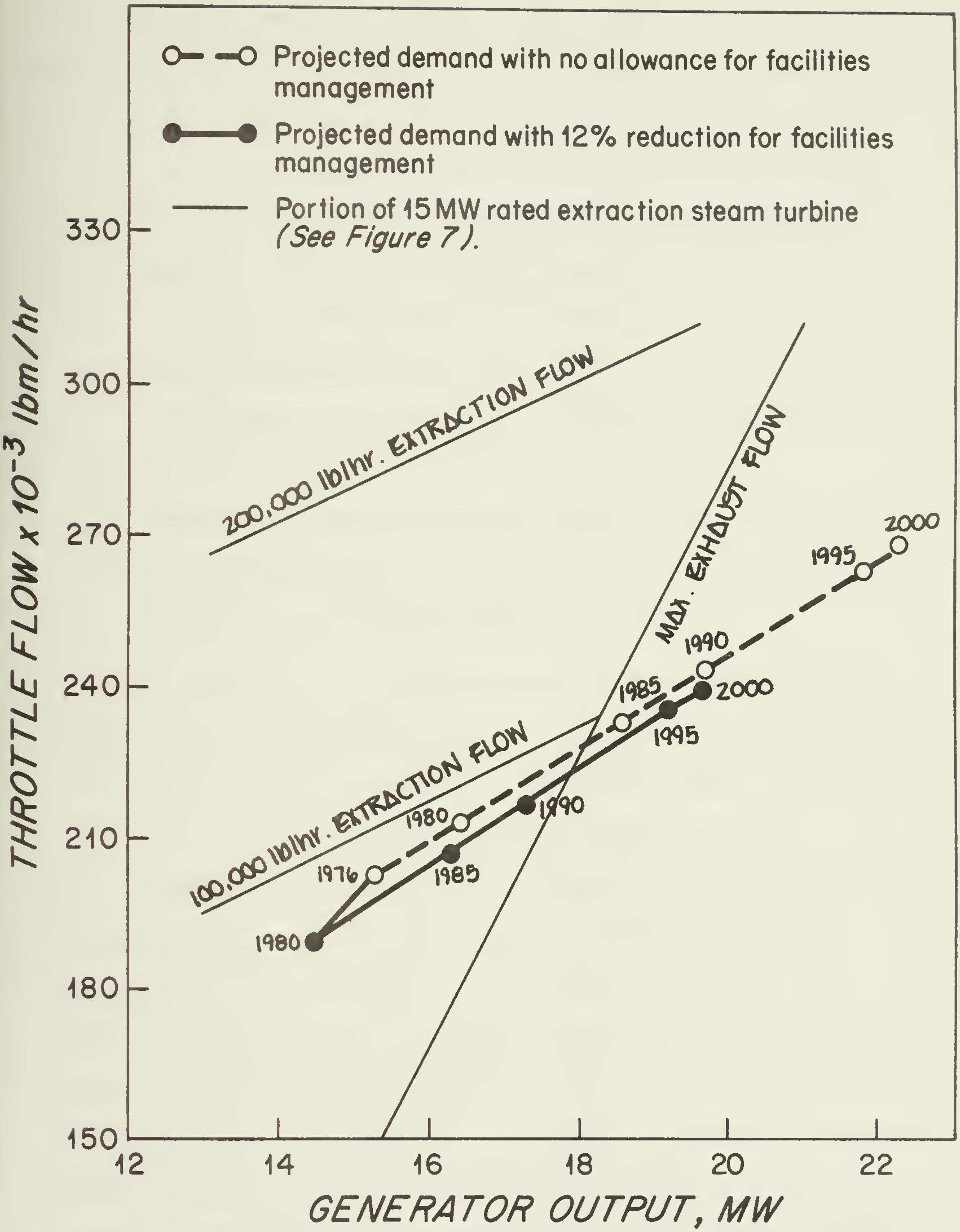

Figure 2.8. Projected peak electrical demand conditions at MIT to year 2000.

1985 period to between 1990 and 1995. As this analysis is based on subjective extrapolation the only conclusions to be drawn are that:

1. The installation of the 15 MW(e) extraction steam turbine unit does not eliminate the possibility of becoming constrained with regard to the portion of the campus load that can be supplied.
2. The alternatives discussed above for the 10 MW unit remain applicable for coping with the limiting condition when the 15 MW unit is employed.
3. Effective load management which include design input into planned construction can control when or even whether the limiting condition is met.

Employment of the steam extraction configuration as a peaking plant is probably not favorable economically due to its higher capital cost and relatively lower operating cost compared to the diesel and gas turbine options, but would present no difficulties or mismatch at the plant/load interface operating with at least a 3 MW(e) base load supplied by the commercial grid. Use of a rated 10 MW steam extraction unit to provide a base load is the better option considering the configuration's operating characteristics, but is not expected to present an economic advantage when the cost of buying peaking power from the commercial grid is considered. Nevertheless this base load configuration would present no additional constraints at the

plant/load interface since with a base load of from 6 to 11 MW(e) the entire range of thermal demand could be supplied.

On the same basis, an installation consisting of two 10 MW(e) units or one 10 and one 15 MW(e) turbine would be able to exactly match all coincident electrical and thermal demands to the year 2000 and beyond. The load matching success of these multiple unit installations suggests the possibility of the phased or imcremental implementation of total energy, for example the initial installation of a small peaking unit followed by the addition of a larger base load plant. Definition of these alternatives will be more obvious upon completion of the initial feasibility study when the characteristics of individual plant performance have been established for the various modes of operation -- peaking and base load.

2.2.2 Gas Turbine Plant

The gas turbine power plant operating characteristics were initially approximated by Was and Mathewson (25) from the characteristics of a General Electric LM - 2500 Marine Gas Turbine for which extensive performance data was available (26). This unit would have rated electrical generation capacity of about 19 MW(e) when coupled with a 97% efficient electrical generator operating at synchronous speed. Considering the exhaust gas parameters for this turbine to be typical of units operating to supply electrical loads from 5 to 20 MW(e), the postulation of a typical waste heat

recovery boiler with 86% efficiency producing 200 psig steam at 420°F led to the formulation of a relationship establishing a gas turbine plant's available thermal power output as a function of the electric power generation. The derivation of this relationship is detailed in Appendix D and the resulting expression is plotted in Fig. 2.9 along with the extreme demand conditions for 1976.

From this figure it is clear that the gas turbine plant with a waste heat recovery boiler cannot by itself satisfy the concurrent thermal and electrical power demands at MIT. Plant configurations based on gas turbine prime movers must include provisions for both dissipating excess thermal energy and supplying additional thermal power generation.

The thermal power production in waste heat recovery boilers is routinely controlled with a bypass damper which diverts a portion of the hot exhaust gas from the turbine around the boiler. This method of control, though it reduces the overall plant effectiveness, does allow the load to be matched when thermal demands are less than the waste heat recovery boiler capacity for the concurrent electrical demand. A more efficient option could be to store the excess thermal energy for use when the thermal demand exceeds that which can be produced from the turbine's exhaust heat. However, the daily load profiles indicate that the variation of thermal demand is much larger with time of year than with time of day, leading to the conclusion that net excess thermal power would be generated daily for a period of at

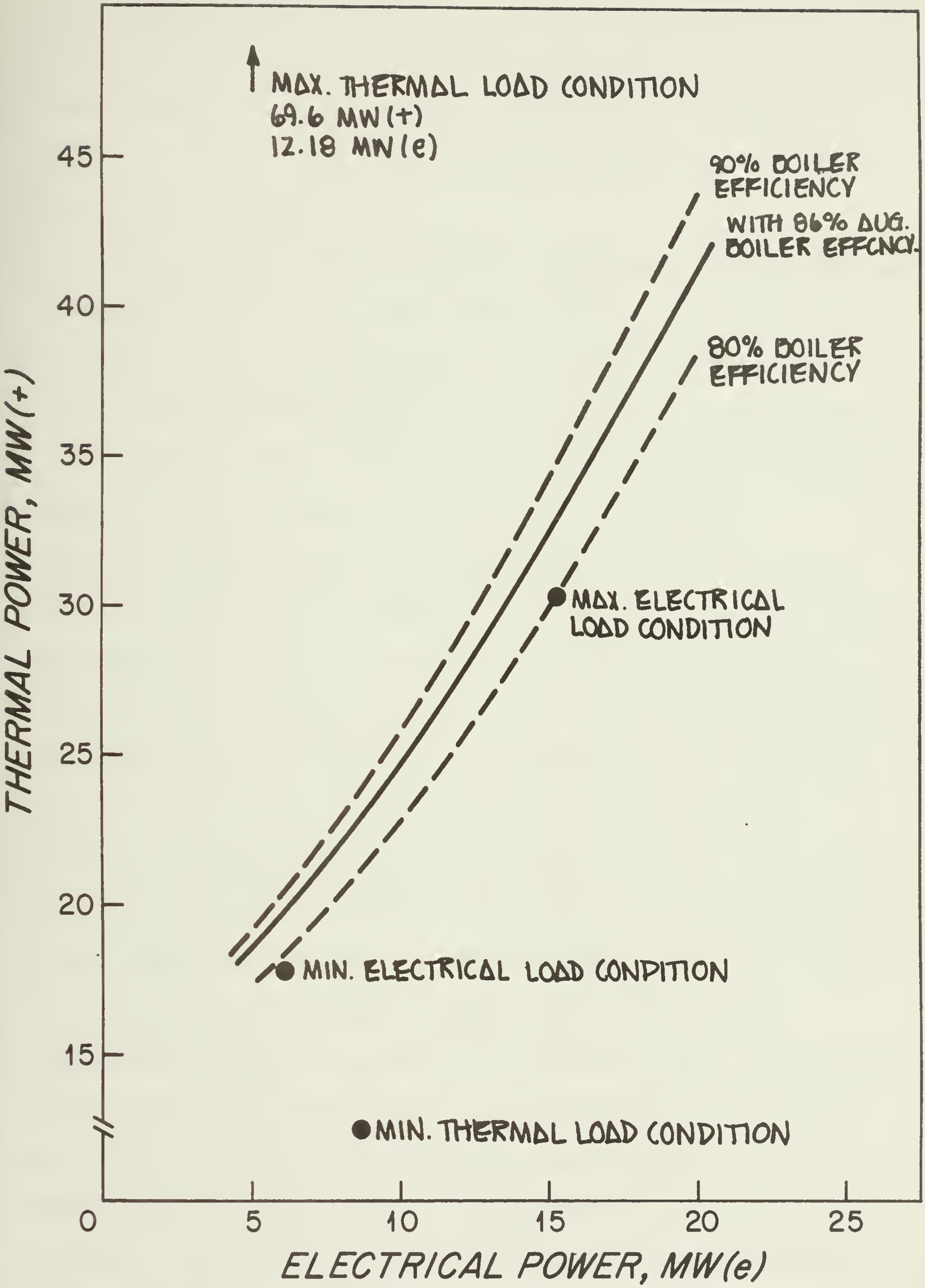

Figure 2.9. Power generation from a 19 MW(e) gas turbine plant with a waste heat recovery boiler.

least several weeks during the temperate periods of the year. Energy storage for these seasonal conditions would require large capacity facilities and , therefore, is probably not feasible at MIT. This will be discussed in more detail in Section 2.3.2 below.

Two options are available for supplementing the thermal power generated in the water heat recovery boiler during periods of high thermal demand. The waste heat recovery boiler can be equipped with burners to add to the enthalpy of the exhaust gases, thus producing higher steam flow rates. These types of boilers are referred to either as supplementary or fully fired waste heat recovery boilers. The distinction here is based on the ultimate temperature on the gas side of the boiler which determines the materials used in boiler construction and hence the cost. Supplementary-fired boilers have gas side temperatures of less than 1800°F while fully-fired waste heat recovery boilers can operate with temperatures in escess of 1800°F. The capacity of both types of fired waste heat recovery boilers is highly dependent on turbine exhaust characteristics: flow rate, temperature, oxygen content, etc. Preliminary information from one vendor (27) indicates that the maximum thermal power available from a supplementary-fired waste heat recovery boiler for the MIT conditions will be about 60 MW(t) when operating with a gas turbine plant generating 16 MW(e). Since this is less than the 1976 maximum thermal demand of 69.96 MW(t) which occurred with an electrical

demand of only 12.12 MW(e), it is likely that a fully-fired waste heat recovery boiler would be required to satisfy the 1976 design load. The more attractive option appears to be the retention and use of a portion of the existing boiler facilities to provide for that part of the thermal demand not satisfied by the unfired waste heat recovery boiler.

Regardless of the boiler options employed, about 40 MW(t) (123,000 lbm steam per hour at MIT conditions) of steam generating capacity must be provided in addition to the unfired waste heat recovery boilers to supply the 1976 demand with a gas turbine plant configuration. From the growth projections, the required additional thermal power generation would be about 45 MW(t) by year 2000, (42 MW(t) if the 12% load reduction from facilities management is realized).

2.2.3 Dual-Fuel Engine Plant

A multiple engine installation was chosen for consideration in the feasibility analysis. This design would employ dual-fuel engines -- engines which, once started on diesel oil, could be shifted over to and operate up to rated capacity on number 6 residual fuel oil. The performance data and heat balance for a typical engine of this type is given in Table 2.2. With the full load electrical generation capacity of this engine established at 6.94 MW(e), a 3 engine installation would be required for the simulation with 1976 demand and a 4 unit design with the projected load

TABLE 2.2

TYPICAL DUAL-FUEL ENGINE/GENERATOR PERFORMANCE DATA

Engine Model Number . . . . Colt Pielstick 18 PC 2U PC 2.3

Full Load Horsepower . . . . 9630 Horsepower

Full Load Generator Output . . . . 6.94 MW(e)

Engine Speed . . . . 514 RPM

Engine Jacket Full Load Temperature . . . 180°F out (20 F° ΔT)

Engine Jacket Cooling Water Flow Rate . . . . 990 gpm

Exhaust Gas Temperature at Full Load . . . . 900°F

Exhaust Gas Flow Rate at Full Load . . . . 132,000 lb/hr

Maximum Permissible Back Pressure . . . . 10" H_2O

Heat Balance - BTU/BHP-hr
(at jacket water outlet temperature constant 180°F)

	% Rated Load		
	50	75	100
Work	2545	2545	2545
Jacket Water	1085	1035	950
Lube Oil	255	247	230
Air Cooler	215	345	510
Exhaust	2178	2064	2220
Radiation and unaccounted for	325	240	155
TOTAL	6603	6476	6610

Exhaust Waste Heat Recovery Boiler: 12,850 lb/hr steam at 200 psig 420°F at engine rated load

Taken from: "Sales Engineering Data Colt Pielstick Stationary Diesel and Dual-Fuel Engines," Colt Industries Fairbanks Morse Engine Division, Beloit, Wisconsin September 1976.

growth to year 2000.

No information was available on the thermal power generation performance of the waste heat recovery boiler -- dual-fuel engine combination at other than rated load. Initial calculations for evaluation of conditions at the plant/load interface were based on a constant 67% waste heat recovery effectiveness (i.e., 67% of the exhaust heat from the engine is converted to thermal energy in the form of 200 psig 420°F steam), and with engine exhaust heat recovery as the <u>only</u> thermal power generation. The 67% value was determined from performance at rated load. These calculations are detailed in Appendix E. The plant operation was postulated to match electrical power generation with demand, assuming equal sharing of the load between the operating units. This is the normal mode of parallel operation with identical synchronous machines. With these assumptions the dual-fuel engine plant's thermal power output was determined as a function of electrical power generated and number of units in operation. The tabular results are included in Appendix E and plotted in Fig. 2.10.

When the dual-fuel engine plant performance curves of Fig. 2.10 are compared with the extreme demand conditions, it is obvious that this plant alone can never satisfy both the thermal and electrical loads. The demand conditions with the minimum thermal load are shown in the figure. All other extreme conditions are higher, off the thermal power output axis scale. Notice that even if the minimum thermal

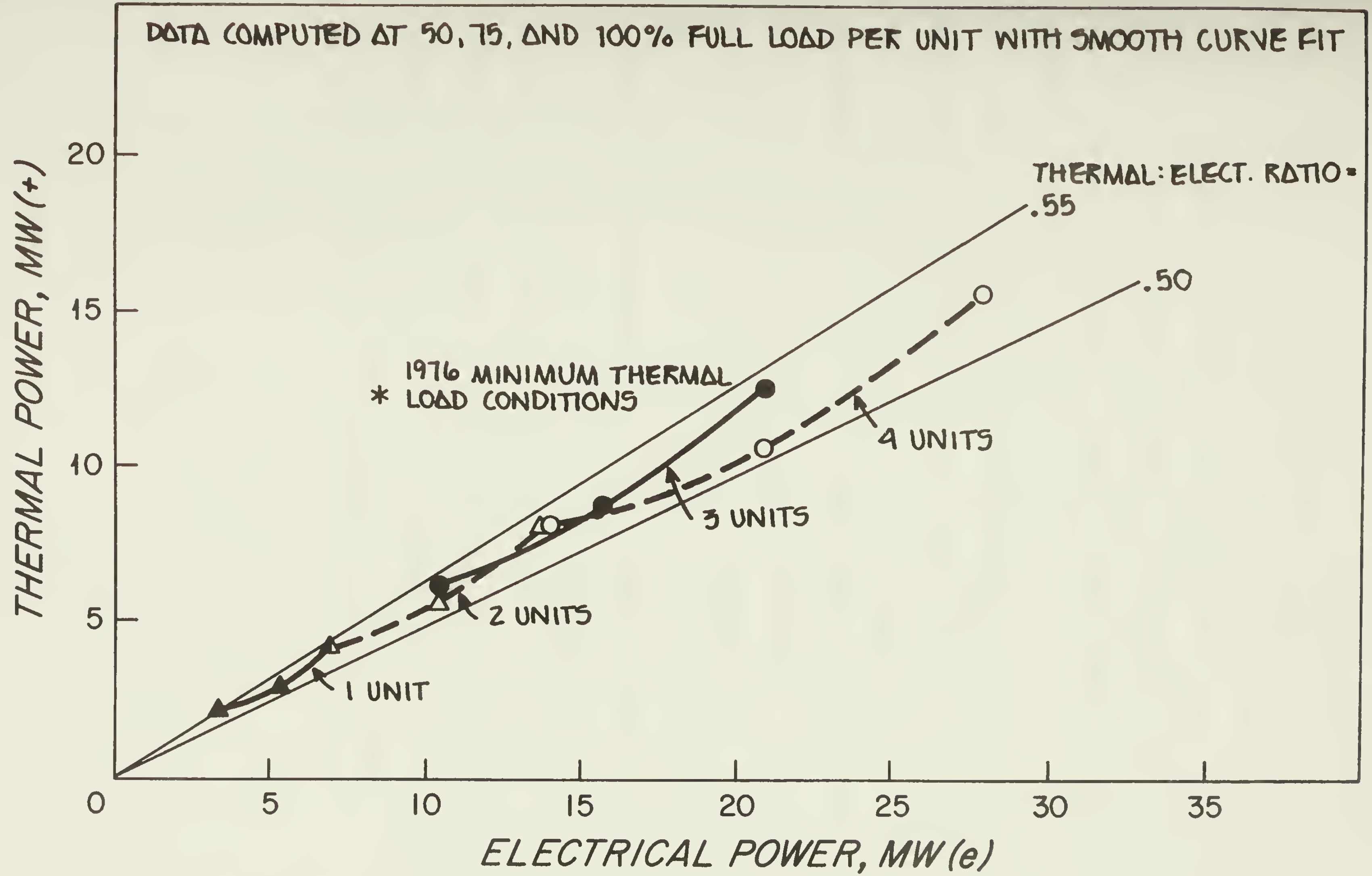

Figure 2.10. Performance characteristics for dual-fuel engine total energy plant.

load occurred at the same time as maximum electric demand of 15.24 MW(e), the plant's thermal power output would be less than demand by over 3.5 MW(t). Since projected plant operation cannot satisfy this unrealistic load condition with its more favorable thermal to electrical load ratio it is fair to say that it cannot satisfy any of the realistic demands with their greater ratios.

Thus, the power plant configuration using dual-fuel engines must include a thermal power source in addition to exhaust heat reocvery. This thermal power source could be supplementary or fully-fired waste heat recovery boilers, or separate boilers as are presently installed. The option selected must be sized to provide thermal power output equal to the maximum difference between thermal demand and the thermal power generated from the engine exhaust. Based on 1976 load data this would require approximately 60 MW(t) of additional thermal power generation. With the growth projections, 72 MW(t) additional thermal power generation could be required by year 2000 (about 64 if the 12% load reduction from facilities management is realized).

2.3 Thermal Energy Storage

In the combined generation of electricity and heat with a total energy plant the output ratio of the two energy forms can be varied only within certain relatively narrow limits; however, the demands for electrical and thermal power may vary independently of each other. As was shown above in the Matching of a specific power plant's generated output to the demand at MIT, instances exist with both the steam extraction turbine and gas turbine plants where the thermal power generated exceeds the demand when the plant is operated to supply the electrical load. These same plant configurations were shown to operate or be capable of operating with a thermal output less than the demand for the concurrent electrical load at other times. Thus, with plant control such that electrical power generation corresponds to demand at all times, some decoupling system is required between the plant's thermal power output and the load. The decoupling system would be required, in the case of the gas turbine plant, to both dissipate and provide thermal energy at different times. The ideal system would be one which, rather than dissipating thermal energy during periods of excess generation, stored this energy for use during periods when the generated thermal power was inadequate to supply the demand. This thermal energy storage system would also be applicable to the extraction steam turbine plant since the extraction steam flow could be reduced below the demand to allow the stored energy to be used during the periods of

of lower electrical loads when the minimum extraction flow was not limited.

The goal of the study of thermal energy storage was, then, to translate this idealized application into a practical conceptual design for incorporation in the MIT total energy system. Since the MIT installation was not to be an experimental facility consideration of storage system designs was limited to those which had been demonstrated to be practical on at least an experimental basis previously.

2.3.1 Review of Energy Storage Applications

Initially the consideration of energy storage was not limited to thermal energy since the instances of thermal supply/demand mismatch could be converted to electrical mismatches simply by revising the plant control to follow thermal demand. Thus, a case where excess thermal energy was generated when the plant was operated to supply the electrical demand would become an instance of insufficient electrical power generation when the plant was operated to exactly match the thermal load.

Storage of energy in a form other than thermal which could later be used for electrical power generation was quickly eliminated as an alternative due to the MIT site constraints. Pumped hydro storage schemes utilize large volumes of water and storage reservoirs with significant static head in addition to requiring plants with hydroelectrical energy conversion capability (28). Compressed

air storage requires large storage volume and has been shown to be economical only when natural storage volumes, such as underground caverns, exist at the site (29). Other storage schemes including the use of such devices as batteries and flywheels have not been demonstrated to provide efficient storage for the magnitudes of energy stored and the time spans of storage which would be required at MIT. There have been practical applications of thermal energy storage in conjunction with total energy systems situated in urban areas.

The concept of thermal energy storage has been developed to meet one of three possible situational requirements. Since the intended function of the storage system is different for each case it is not surprising that radically different system design configurations have been applied. The three different purposes for the storage and the resulting configurations are:

1. Peak shaving. The storage facility provides thermal energy during periods of high demand with the energy of the storage reservoir replenished during the periods of low demand. In this manner, storage is utilized to reduce the magnitude of load variation resulting in a reduction of the required generating capacity and an increased plant load factor. The storage reservoir is routinely located at or nearest the energy use point reducing the size of the distribution line since it need not be sized for the

peak demand.

2. Supply in conjunction with intermittent sources. A portion of the thermal energy generation plant's output is stored while the plant is operating and then utilized as a supply while the plant is shut down. Solar energy plants are the major users of these systems and the reservoir is normally located near the generating site.
3. Reserve supply. The storage installation provides a ready reserve for short interruptions in generating capability. The economy of scale for this use generally favors a reservoir at the generating site, though there have been successful systems with reservoirs at the consumption or use point (30).

This division of thermal energy storage applications is not completely distinct since systems designed for peak shaving or use with an intermittent source could fulfill the reserve supply requirement when they are charged. The division does point out the different design motivations and leads to consideration of the time scales involved in the storage.

Systems designed to provide a reduction in the peak thermal demand seen by the plant are, of course, designed to have capacities equal to the energy (integrated power demand over time) reduction sought and to provide storage over the time from minimum to maximum demands. The magnitude of time and energy involved depends upon the demand profiles. Shaving annual peaks requires very large capacity systems

and storage capability on the order of 6 months whereas reduction of daily peaks could be accomplished with a smaller capacity system and 12 hour storage. Applications for peak shaving on a daily basis are very similar to the designs for use with intermittent solar (diurnal) sources. Reserve supply applications are usually considered to be very short time storage supplies, on the order of an hour or less. The application for the MIT system is actually a load matching situation resembling both peak shaving (in reverse, since energy production peaks are treated as opposed to demand peaks) and operation with intermittent sources. The storage reservoir <u>must</u> be charged during periods when the plant thermal output exceeds demand, as with peak shaving, or the reservoir can be treated as being charged from an intermittent source equal to the supply/demand mismatch. The distinction is really irrelevant since the design decision of where to situate the reservoir(s) is determined from the facts that at MIT:

a. the distribution lines are established and will not be modified in the forseeable future; and

b. the individual buildings' thermal demands vary over a larger range than the entire campus load.

This eliminates consideration of distribution line sizing and indicates that individual building storage units would require a larger total capacity than a centralized storage reservoir. This, combined with the economy of scale favoring

large reservoirs, leads to the conclusion that thermal energy storage at MIT should utilize a central storage reservoir.

Several important research projects have been conducted to determine practical thermal energy storage schemes. These studies (30 to 35) have included comparisons of the various materials which could be used for energy storage, as well as specific design configurations. A review of this work has led to the conclusion that present applications of thermal energy storage with artifical reservoirs should employ a liquid as the storage medium, storing the energy in the form of sensible heat. Liquids characteristically exhibit large heat capacities per unit volume and thus storage systems using liquid storage media require smaller volume reservoirs than those employing solids, gases, or phase change materials which store energy primarily as latent heat. Steam storage has been used for many years but only on a short time span, small capacity basis (36).

The choice of the liquid for the storage system is strongly coupled to the fluids and the states of these fluids which are available from the thermal power generating source and which are to be used in the distribution network. Water is the preferred liquid used in all existing thermal energy storage systems with large central reservoirs. Some special heat transfer oils have been developed which have advantages over water for energy storage. However, their cost, the heat-exchanger requirements, and the operational

safety problem with their use have eliminated their consideration for large scale centralized storage systems (37).

The most impressive operational applications of thermal energy storage in configurations similar to that envisioned for use at MIT are found in Europe. These applications are in conjunction with heat and power (total energy) plants serving large district heating systems (38). One of these type installations in Malmo, Sweden employs four high temperature water storage tanks, each with a volume of 2500 cubic meters (about 8800 ft^3) with a storage capacity of about 155 MW(t)-hr equivalent to 5 hours of the plant rated thermal output. These tanks are charged with hot water (approximately 250°F) at the top, displacing cooler water (about 150°F) from the bottom of the tank into the return line from the district heating system during the periods when the thermal power output of the back-pressure turbine electrical generation facility exceeds the heating system's demand. The interface between the hot and cold water, called the thermocline, moves up or down in the tank indicating the charge level. The tanks are charged and discharged routinely on a daily cycle. There has been no reported difficulty with the diffusion or spreading of the thermocline, probably due to the large height to diameter ratio, about 5:1, of the tanks. Insulation of the tanks has reduced heat loss to about 10% in one week (though the tanks are normally discharged sooner with less than 3% loss of the heat stored in a daily cycle), (39). It must be noted that

this successful application of thermal energy storage is in conjunction with a high temperature water heating system, and thus not directly applicable to MIT with its steam heating.

Where steam is used as the thermal energy distribution medium, the advantages of thermal energy storage in the form of hot water may be incorporated into the system by storing pressurized hot water. The storage reservoir is then discharged to the steam distribution main through a pressure reducing valve allowing a portion of the hot water to flash to steam at the lower pressure. This type of system relies on the availability of energy at high temperatures to produce the high temperature water for storage. One disadvantage of this system is that only a portion of the storage volume is actually used to store energy which will ultimately reach the distribution network. For example, to get 200 psi steam from such a system, the latent heat of vaporization would be acquired from a reduction in the energy of the remaining water in the storage reservoir. Thus, with hot water stored at 1000 psia isentropic expansion to 200 psi would result in about 22% of the water being flashed to steam with the remaining 78% of the water at saturated conditions for 200 psi. This isentropic expansion process is shown in the temperature-entropy (TS) diagram of Fig. 2.11. In this diagram, point D is the state for stored saturated water at 1000 psi and point B is the state of the stored fluid after the pressure reduction from 1000 psi. The state at point B

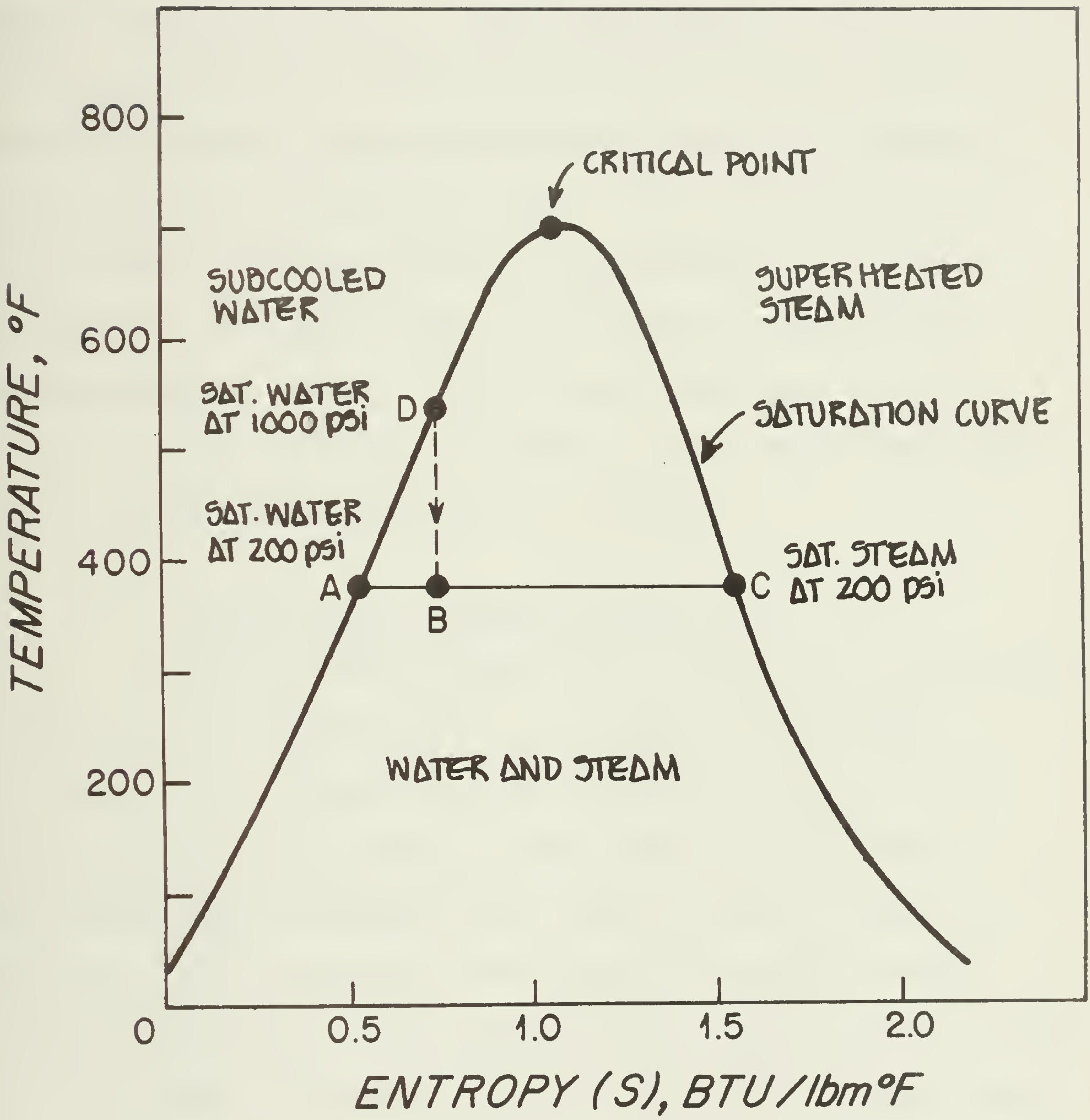

Figure 2.11. Isentropic expansion of stored hot water to produce steam at 200 psi.

is composed of a mixture of water and steam at states denoted by points A and B respectively. The fraction of steam in this mixture is computed from the length of line segment AB divided by AC. Even with storage at the critical pressure, expansion to 200 psi would only flash about 66% of the water to steam and would require a reservoir designed for greater than 3000 psi pressure. Thus, two conditions are necessary for this storage configuration to be useful: the energy to be stored must be available in the form of high temperature water, and the water remaining in the reservoir after the pressure reduction and steam discharge must be employed effectively in the system.

2.3.2 Evaluation for Application at MIT

Thermal energy storage was considered for use with a total energy system at MIT in a number of different configurations. Mating of either the gas turbine plant or the extraction steam turbine plant with the MIT demand profiles indicated periods when excess thermal energy was available. The dual-fuel engine plant exhibited no such characteristic, i.e., its thermal output would always be less than the demand. There is, then, no need for thermal energy storage in conjunction with the dual-fuel engine plant design. Employment of thermal energy storage with each of the other basic plant designs was considered separately since each plant presented a different potential and different constraints for energy storage.

When matched to the 1976 load the rated 10 MW(e) extraction steam turbine plant provided excess thermal energy during some periods, but the 15 MW(e) plant did not. The larger plant would only reach this condition with demands projected beyond 1990. As the feasibility analysis was to be based on supplying the 1976 demand profiles, thermal energy storage should not be applied to the larger plant design. Additionally, effective load management would eliminate the excess thermal energy supply condition entirely by establishing a thermal-electrical load mix which, with demand growth, could be within the capability of the 15 MW(e) plant if it were installed.

An hour-by-hour matching of the 10 MW(e) extraction steam turbine thermal output to the 1976 hourly demand showed that the total yearly excess thermal power production was about 6750 MW(t)-hrs. This is the maximum power which could be conserved with thermal energy storage, and represents only 2.5% of the total yearly thermal power demand. The overproduction occurred for a maximum of 7 hours per day with some excess thermal production on only 96 days, all in the period from April to October. These figures indicate that a thermal energy storage system with this plant should be sized for daily cycling, but would be utilized only 27% of the time -- a low utilization factor.

The form of the excess energy available for storage with the extraction steam plant is the most disadvantageous factor in attempting to employ thermal energy storage with this

plant configuration. The energy available to be stored is necessarily in the form of 200 psi, 420°F extraction steam. Storage of the maximum daily thermal power overproduction, 111.48 MW(t)-hr, as steam at extraction conditions would require a steam storage volume of about 800,000 cubic feet. With no natural storage volumes at the MIT site this volume of storage would require the expensive construction of large pressure tanks occupying more space than is available.

The heating of feed or makeup water with the excess extraction steam is a possibility. In heating the condensate return from an average temperature of about 165°F, direct feedheaters could utilize approximately 0.03 pound of excess steam per pound of feed water heated. During the maximum 1976 mismatch period an average of about 2880 pounds per hour of excess extraction steam could be utilized in this manner. This is about 6.5% of the total excess. Use of all the excess thermal power to heat feed and makeup water for use as required would avoid the steam storage difficulty. With about 8% makeup required by the steam distribution system and the makeup water available from Cambridge water supply at an average of about 70°F, approximately 0.1 pound of excess steam could be used per pound of makeup water heated to 210°F. During the maximum 1976 mismatch period makeup would be heated at an average rate of about 50,000 gallons per hour above the usage rate for 6 hours or 300,000 gallons of make-up to be stored requiring a storage volume of 46,000 cubic feet. This amount of stored makeup feed would be sufficient

to supply the plant for over 300 hours at the yearly maximum makeup rate -- much in excess of that required for daily cycling of the storage system. Heating of feed and makeup water should be considered in the plant design but not from a thermal energy storage point of view. The fact that thermal energy is available only in the initial form of steam and must ultimately be utilized as steam at similar pressures and temperatures makes practical use of thermal energy storage impossible with the 10 MW(e) extraction steam turbine plant.

Hourly comparisons of the potential thermal power which could be generated by waste heat recovery from the 19 MW(e) gas turbine exhaust with the 1976 thermal demand indicates a potential for long term thermal energy storage. Fig. 2.11 shows the results of this comparison with a bar graph of both the number of days per month on which excess thermal power could be produced for only a portion of the day and the number of days per month during which excess thermal power is available during the entire day. With the previous observation that the magnitude of annual thermal demand variation is larger than that of the daily variation at MIT, Fig. 2.11 may be interpreted as showing that a short term thermal energy storage system could be fully charged and discharged on a daily basis during only three months of the year (April, May, and October), would remain at or near full charge for months (June through September), and would be fully discharged for the remaining five months. Long

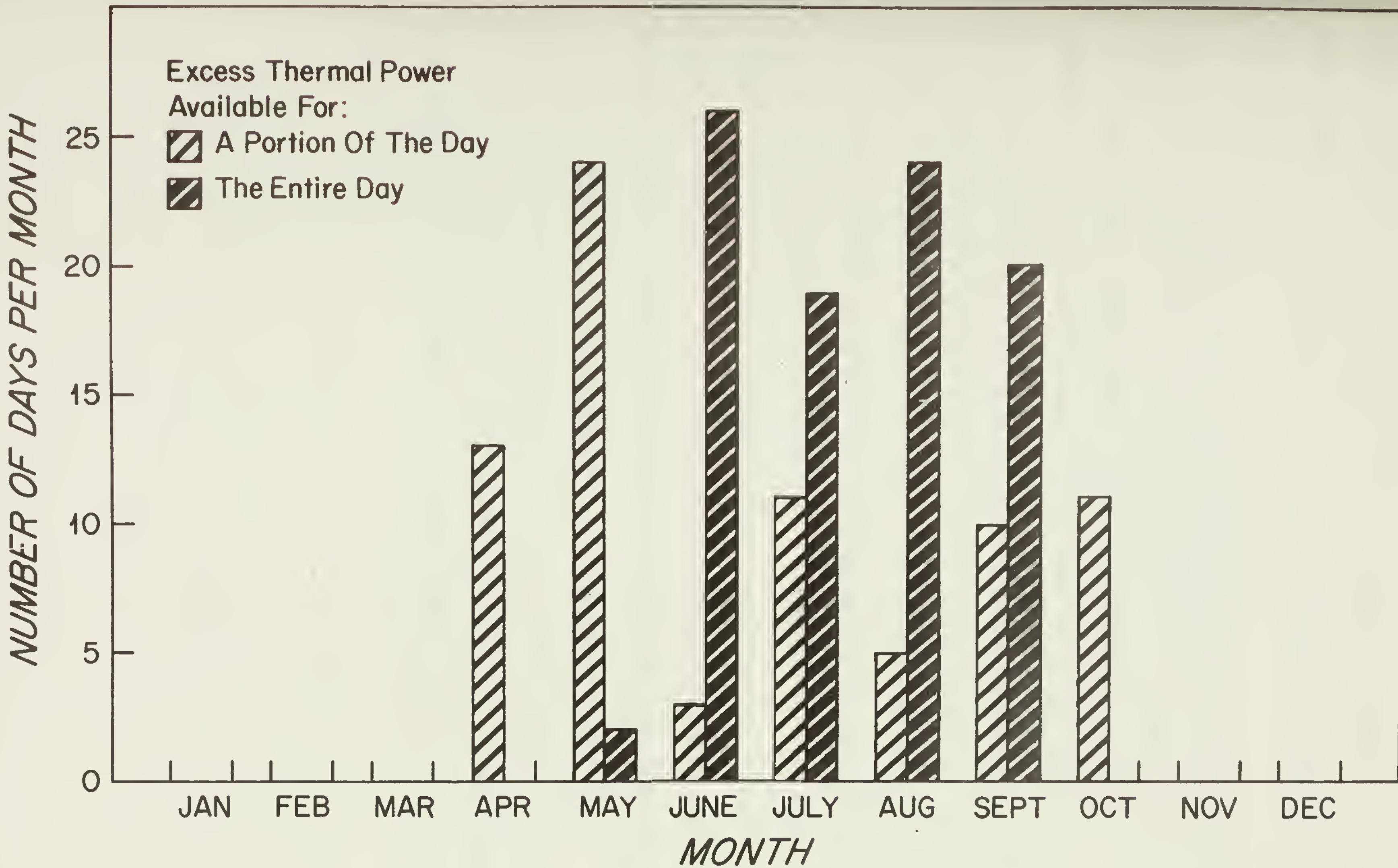

Figure 2.12. Number of days per month on which excess thermal power would be available with 19 MWe gas turbine supplying 1976 MIT demand.

term or seasonal thermal energy storage is just not practical with anything other than natural storage reservoirs.

The fact that with both plants there is excess thermal power available only during the period from April to October indicates that tailoring of MIT's air conditioning demand could result in more efficient utilization of either of these plants. The replacement of electrically powered air-conditioning units with central chilled water cooling or absorptive units, both of which use steam, would serve to both decrease the electrical demand and utilize more thermal power during the period when the increased electrical power demand from the total energy plant resulted in excess thermal power production. If either the 10 MW(e) extraction steam turbine plant or the gas turbine configuration are shown to be competitive in supplying the 1976 demand, the load projection for the design life of the plant should include a study of the MIT air-conditioning system and recommended tailoring of this load to be more compatible with the plant's capability.

III. PLANT/ENVIRONMENT INTERFACE - WASTE HEAT DISSIPATION

Consideration of the plant/environment interface for the final and detailed design of a total energy system must include a projection of the environmental impact of the system, i.e., a determination of the sound or noise, the exhaust products, and the heat which will be emitted by the power palnt. Equipment must be included in the design for the abatement or control of these emissions to within acceptable standards. From the viewpoint of the system operation which is to be considered in a feasibility analysis by simulation, most of the equipment employed to modify the system's environmental impact is relatively passive; the envioronmental protection equipment does not significantly affect the performance of the system. This is the case with sound proofing or exhaust particulate removal. The cost of this equipment can be included in the system capital cost without detailed design and modeling of its operation so long as the requirements for its use are within the capability of available equipment. The means of waste heat dissipation is, in the same sense, active, with the potential for having a significant effect on the operation of the system; therfore, design and performance modeling of the waste heat dissipation equipment should be studied for inclusion in the simulation portion of the feasibility analysis.

3.1 Plant/Environment Interface Constraints

Determination of an optimal waste heat dissipation scheme for each plant alternative is initiated with the definition of the constraints imposed by the site selected for construction of the power plant and by each of the plant configurations.

3.1.1 Site Constraints for Waste Heat Dissipation

The initial selection of the site for construction of a power generation facility should be the result of a detailed study in itself; however, when the facility is to be a total energy installation serving an existing complex, energy distribution considerations require that the sites considered be limited to those within close proximity to the complex. Furthermore, when the complex is being served by an energy distribution network which is to be retained for use with the new installation the site should be in an area where the plant can be connected to this distribution system effectively. At MIT the existence of a central heating and cooling plant, portions of which are to be retained in the new system, mandates that the site be considered as an extension of the existing plant. This area, shown in Fig. 3.1, provides oil storage facilities, chill water equipment, and the campus central heating and cooling controls installation as well as access to the 200 psi steam heat distribution network. The existing installation also contains the major electrical transformer supplying the campus with electrical power from

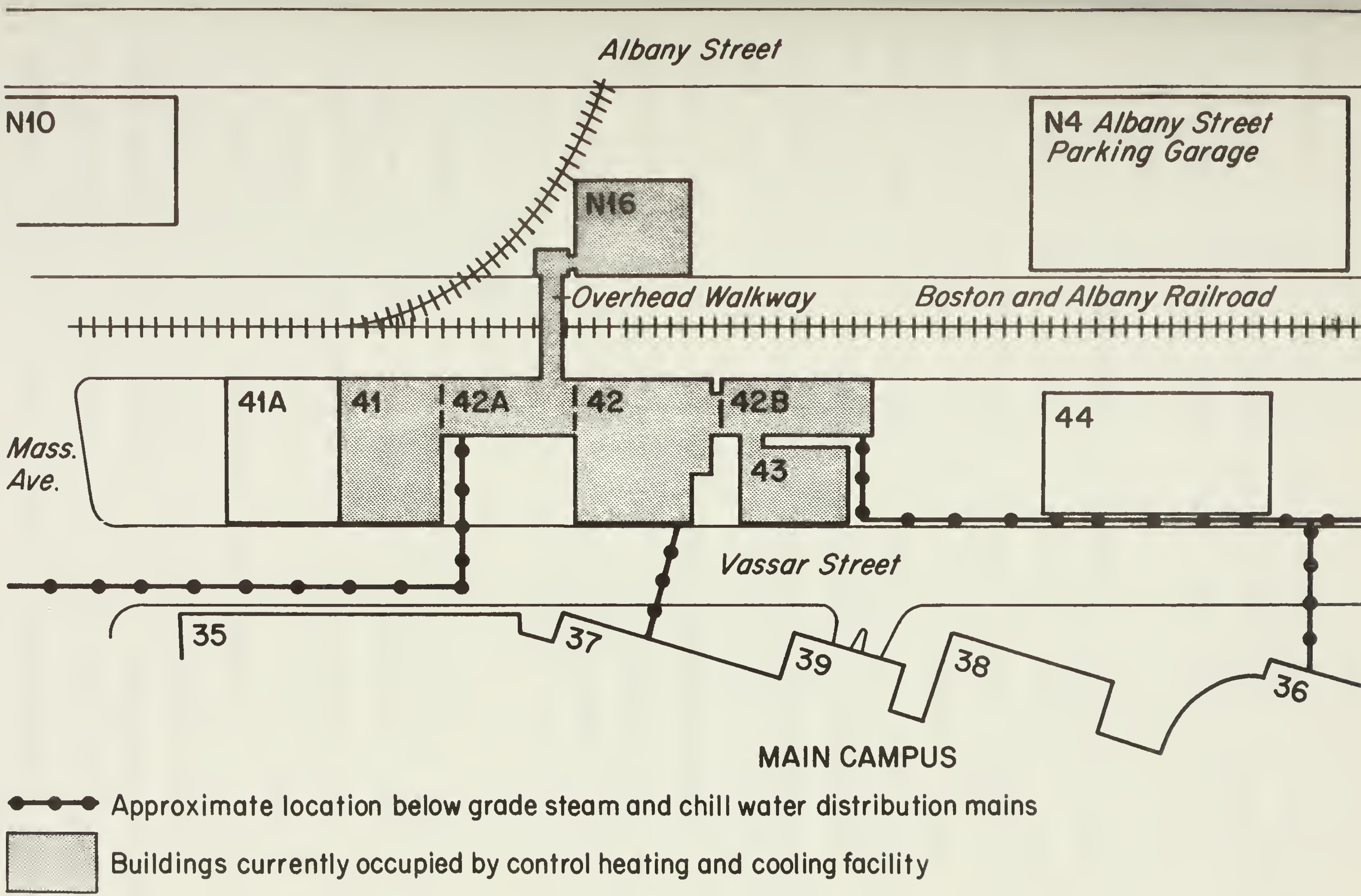

Figure 3.1. MIT's central heating and cooling facility.

the commercial supply grid. There is sufficient land adjacent ot the existing plant for construction of buildings to house the proposed facility and there is a possibility of installing the total energy plant within the existing structures.

The site is approximately 1600 feet from the bank of the Charles River, which could provide a source of cooling water for the proposed plant; however present indications are that such use of the river would be opposed by the Metropolitan District Commission which is the state agency controlling use of the waters and lands of the Charles River Basin area (40). Though present legal minimum requirements do not prohibit use of the river water for cooling (41), consideration of the protracted legal process and the extensive design for protection of the river ecology which would be required prior to use of the river water necessitates exclusion of the Charles River as a source of once-through cooling in the feasibility study.

Cooling water for the existing chilled water facility (condenser and chiller cooling) is provided by 4 two-cell, incuded-draft counterflow cooling towers installed in groups of 2 towers (4 cells), each on the roofs of buildings 42 and N 16 (see Fig. 3.1). Cooling towers 1 and 2 on the roof of building 42 are wooden towers with a design waste heat dissipstion rate of 28.12 MW(t). Tower 3 adjacent to tower 4 on the top of building N 16 is a steel tower with design dissipation of 32.81 MW(t). Tower 4, also of steel construc-

tion, has a design dissipation rate of 37.26 MW(t). The design conditions for all towers are 117°F inlet, 85°F outlet at 76°F wet bulb. These cooling towers are adequate for the existing chilled water facility. They also provide empirical data and experience which can be used as a basis for comparison of similar cooling tower designs for use at MIT.

Cooling towers 3 and 4 were originally designed and constructed as the first two increments of a possible ultimate installation of 12 cells. This total proposed installation is sketched in Fig. 3.2. The service to be provided by the possible construction of 8 additional cells in the building N 16 area has not been specifically allocated though the original consideration was to provide cooling for an expanded chilled water facility. This location is available for installation of waste heat dissipation equipment to serve the total energy installation.

Experience with the wet cooling tower configuration at MIT has been entirely favorable with little or no difficulties in operation. There has been no problem with visible plume behavior in the area and only one initial complaint of mist in a parking lot adjacent to building N 16 (42). This parking lot is no longer extensively used and, with the improved methods for carry-over control which have been employed in cooling tower design since the last installation at MIT in 1974 (43), mist or carry-over would probably not represent a problem. These site considerations, then,

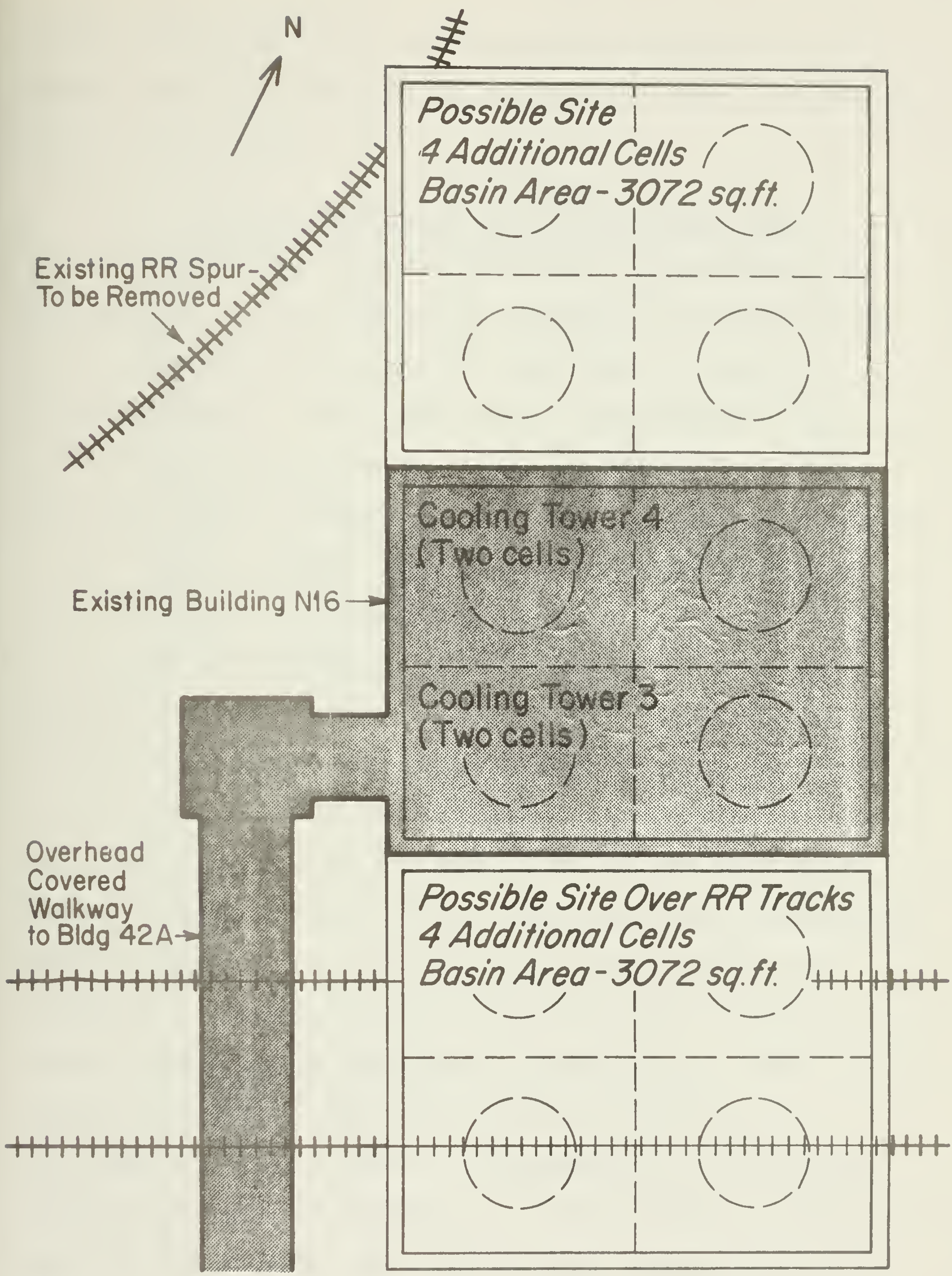

Figure 3.2. Possible site for additional cooling towards at MIT.

dictate the employment of some type of cooling tower as the primary waste heat dissipation method when water is utilized as the cooling fluid.

3.1.2 Extraction Steam Turbine Plant Waste Heat

Waste heat from the extraction steam turbine installation is in the form of the latent heat of condenation of the turbine exhaust from the last stage of the low pressure section. When this energy is removed with a conventional water-cooled condenser at the turbine design operating exhaust pressure of 3.0 inches of mercury absolute, it will amount to about 1031.4 BTU's per pound of exhaust steam. This value is determined from the difference between the exhaust steam enthalpy of 111.4 BTU/lbm at atmospheric pressure and 112°F, which allows for 3 F° condensate subcooling below saturation temperature at 3 inches Hg to provide a net positive suction head for the condensate pumps. With this value of specific waste heat and the performance curves of Figs. 2.4 and 2.7 the total waste heat dissipation capacity required can be calculated. These results are presented in graphical form in Figs. 3.3 and 3.4 for the rated 10 MW(e) and 15 MW(e) units respectively. These figures indicate maximum rejected heat rate values of 28.29 MW(t) and 41.61 MW(t), corresponding to the maximum exhaust flow rates of 93,600 and 137,000 pounds of steam per hour for the 10 MW(e) and 15 MW(e) units respectively. The calculations presented in Section 2.3.1 confirm that this

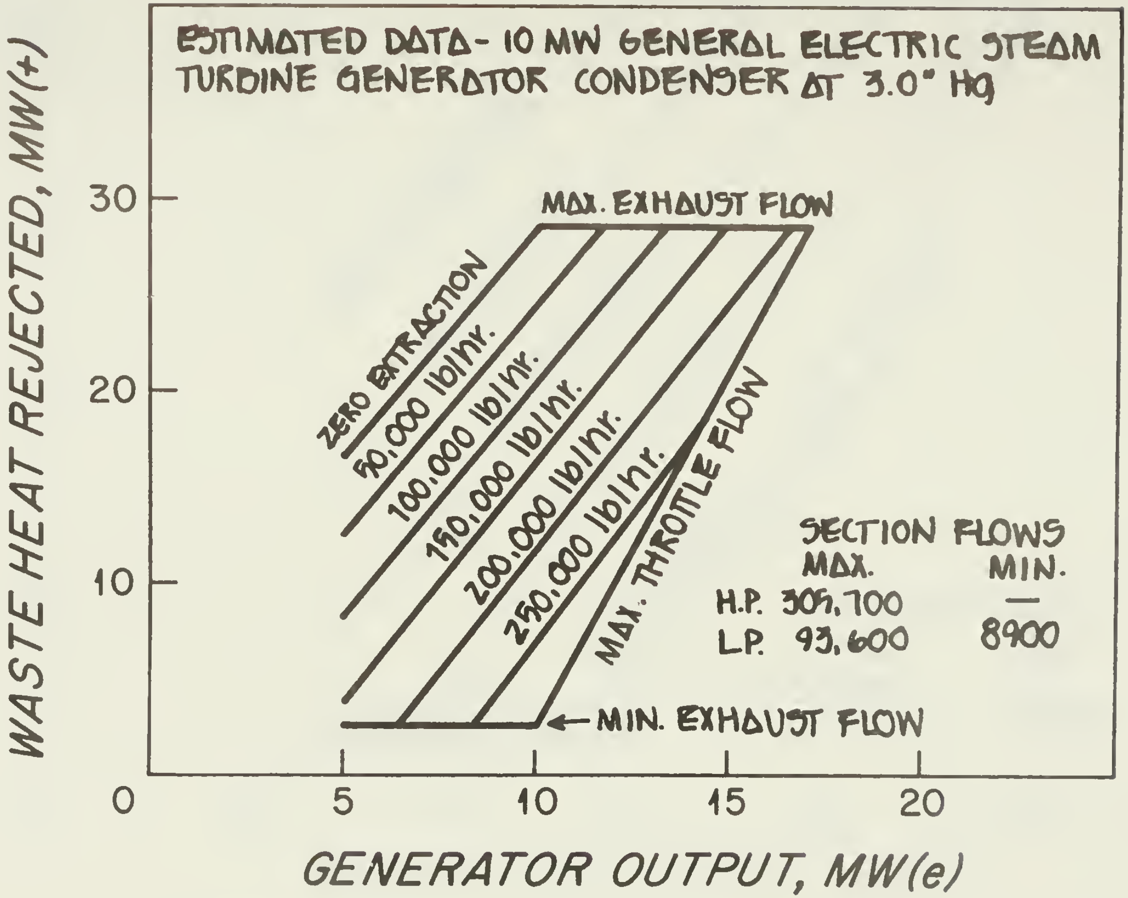

Figure 3.3. Waste heat rejection from a 10 MWe extraction steam turbine installation.

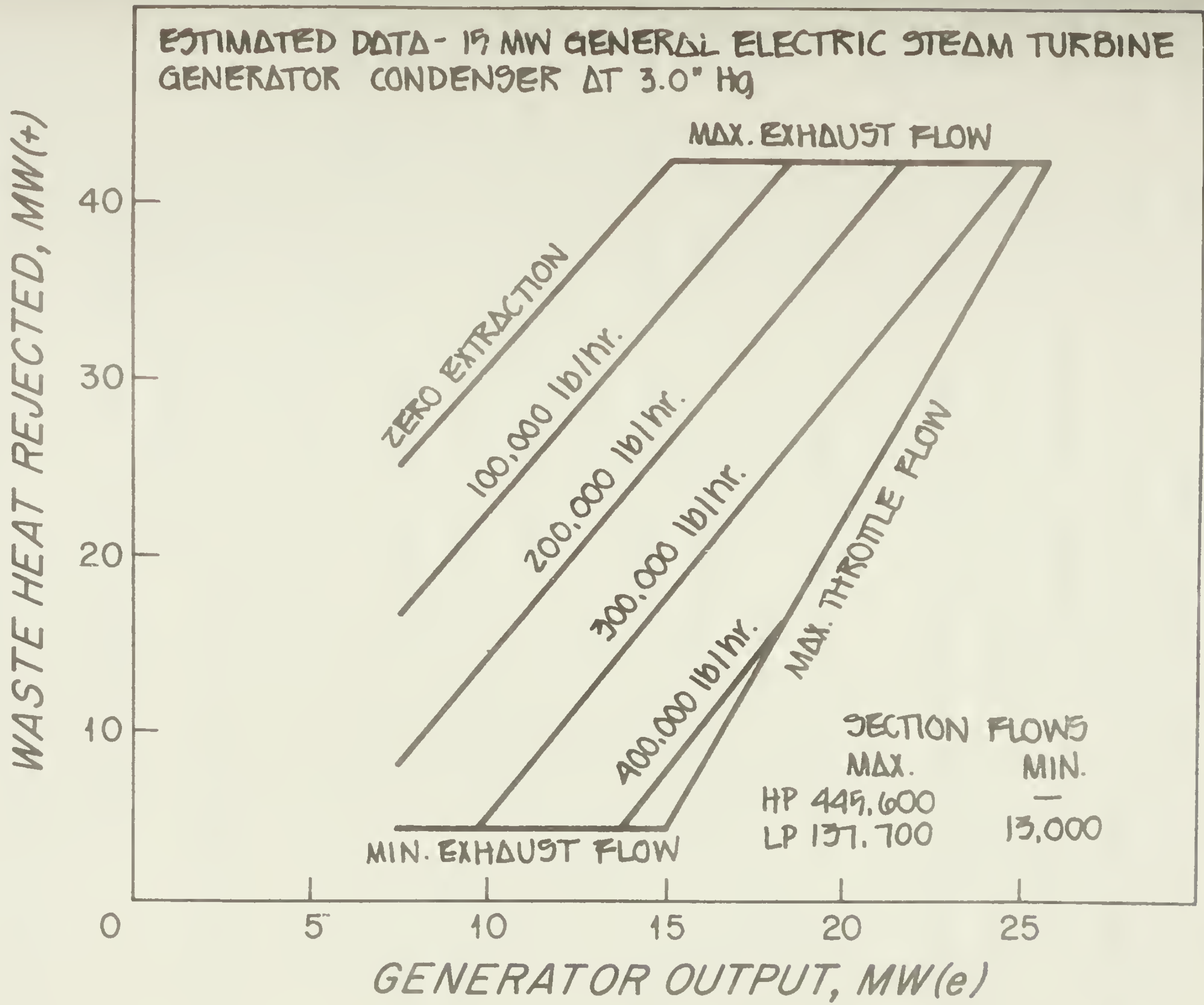

Figure 3.4. Waste heat rejection from a 15 MW(e) extraction steam turbine installation.

maximum heat rejection condition is reached with operation of the 10 MW(e) unit supplying the 1976 load, and would occur some time prior to the year 2000 with current load growth projections. To avoid further limiting an extraction steam turbine plant with inadequate waste heat dissipation for its maximum generation capacity, the waste heat dissipation facility should be designed to have a capacity equial to the heat which must be removed from the maximum exhaust steam flow.

The extraction steam turbine specifications limit the maximum exhaust pressure to 3.0 inches of mercury for design full load operation. With saturation conditions in the condenser, maximum exhaust temperature must then be 115°F The condenser is not supplied by the vendor as part of the turbine unit and may, therefore, be considered to be a portion of the waste heat dissipation system. The object of the optimization of design parameters for the waste heat dissipation system is, then, the determination of the:

a. cooling water flow rate;

b. cooling water temperatures at the inlet and outlet of the condenser; and

c. condenser heat transfer surface area,

which will be required to provide condensation at 115°F for the turbine's maximum exhaust steam flow.

3.1.3 Gas Turbine Plant Waste Heat

The greater part of the waste heat from the gas turbine plant is contained in the exhaust gases. The portion of this heat not recovered in the waste heat recovery boiler is exhausted directly to the atmosphere. A small portion of the total waste heat is dissipated by radiant and air convective cooling of the turbine-compressor unit and its associated piping. This waste heat will be a factor to be considered in the complete design of the facility, especially in determining the ventilation requirements in the building where the turbine is located, but it has negligible effect on the initial conceptual design. The only other waste heat in the gas turbine installation is the heat which is removed from the turbine-compressor unit via the lubricating oil. To maintain the oil's lubricating quality, the energy or heat which has come from the frictional heating of the bearings must be removed and the oil must be maintained within a given temperature range.

The most effective oil cooling and that usually specified by gas turbine manufacturers is achieved in oil water heat exchangers which, of course, require cooling water flow. For the MIT installation, based on the characteristics of the General Electric LM-2500 Gas Turbine (26), oil cooling requirements would be about 5,800 BTU/min with oil temperature out of the cooler from 150°F to 170°F and maximum oil flow of 17 gal/min. The specific cooling water requirements will depend upon the design of the oil-water heat exchanger

selected or provided with the turbine compressor unit. Indications from gas turbine vendors are that these heat exchangers are usually supplied with the turbine compressor unit and require a cooling water flow of about 9 to 12 gpm at 80°F design inlet temperature.

3.1.4 Dual-Fuel Engine Plant Waste Heat

The heat balance for the dual-fuel engine in Table 2.2 indicates that the majority of the rejected heat, approximately 35% at full load, is removed from the engine via the exhaust gases. The portion of this engine waste heat that is not recovered in the waste heat recovery boiler is rejected to the atmosphere at hot exhaust gas. As additional 4% of the total waste heat at fully load is in the form of radiant heat from the installation. This energy will serve to heat the plant area adjacent to the engine and will be a factor to consider in the final, detailed installation design; however, it is of little consequence in the conceptual design and modeling. The remaining waste heat, primarily from kinetic frictional heating and heat conduction from the cylinder and blower, would overheat engine components if it were not effectively dissipated. The engine design contains internal water and oil circulation paths to remove this heat from the engine. Heat exchangers are available to then remove the heat from the primary air and oil to a secondary circulating water system. The jacket cooling water may be cooled directly by the circulating water system or in-

directly with a water to water heat exchanger. Direct supply from a closed loop circulating water is not recommended due to probability of oil and carbon contamination of the circulating water. Effective plant designs have utilized jacket water, air, and lube oil coolers to preheat the feedwater for waste heat recovery, or conventional boilers (19). The economics of these design configurations are considered in the study of the power plant alternatives (44).

The cooling system specifications that could be required for various dual-fuel engine plant configurations can be treated incrementally if the rejected heat is considered on a per engine basis at full load. The waste heat dissipation capability on this basis will be sufficient for design operation of one engine with any plant configuration. Where utilization of waste heat for feedwater preheating is incorporated in the plant design there will be excess waste heat dissipation capacity which will allow complete flexibility in the operating configuration for situations in which boiler operation is not desired or required concurrently with engine operation for electrical power generation.

The basic unit for the dual-fuel engine conceptual designs was selected as Model 18 PC2V supplied by Colt-Pielstick. This engine is supplied equipped with a cooling water pump; heat exchangers; and piping for air, oil, and jacket cooling, with cooling water supply inlet to engine conditions specified. The cooling water inlet specifications with projected full load cooling water outlet conditions are

shown in Table 3.1. The projected outlet conditions are based on the design cooling water supply in series to the air, oil, and jacket cooling, in that order, at full load conditions, with a unity specific heat for cooling water, at the design cooling water pump flowrate. With the engine prescribed by the vendor to include the cooling water circuit, the full-load cooling water requirements are completely specified as 1200 gpm for each engine supplied at maximumum temperature of 85°F with 4.75 MW(t) rejected heat load (112°F cooling water outlet). This presents a more constrained design conditions for waste heat dissipation than the extraction steam turbine configuration where only the waste heat load specified, leaving the cooling water flow rate and temperatures and thus the condenser design as variables for optimization of waste heat dissipation.

TABLE 3.1

COOLING WATER REQUIREMENTS FOR DUAL-FUEL ENGINE

Cooling Water Flow Rate	1200 gpm
Cooling Water Pump Developed Head	30.35 psid (70 ft)
Maximum Cooling Water Temperature to Engine	85°F
Projected Full Load Cooling Water Outlet Temperature	112°F
Full Flow Cooling Water Pressure Drop Across Engine	6.25 psid
Available Head at Engine Cooling Water Outlet	24.10 psi (55.6 ft)

Requirements based on data for Colt-Pielstick Model 18 PC 2U Dual-Fuel Engine as described in "Sales Engineering Data, Colt Pielstick Stationary Diesel and Dual-Fuel Engines," Colt Industries Fairbanks Morse Division, Beloit, Wisconsin, September 1976.

3.2 Waste Heat Dissipation Options

All of the alternative plant configurations require cooling water as the medium for at least a portion of the waste heat dissipation at the plant/environment interface. Site constraints eliminate the consideration of once-through cooling; therefore, a water to atmosphere heat exchange device is required for waste heat dissipation. These types of devices may be considered in two general classifications -- evaporative and closed or non-evaporative. The evaporative devices require a small portion of the circulating water to be evaporated. The evaporation process takes the energy for latent heat of vaporization from the remaining water. The loss of this energy plus the energy lost in conduction from the water to the cooler air results in lowering of the temperature of the circulating water. Makeup water must be added to the circulating water to compensate for the water lost in evaporation. The non-evaporative devices are simply closed, water-air heat exchangers whereby cooling of the circulating water is accomplished by conduction from the water to the air across the separating material (normally metallic finned tubing). The predominant advantage of the non-evaporative device is that no makeup is required.

Due to the large heat transfer surface required, non-evaporative devices normally cost 3 to 5 times as much as the most expensive evaporative system for the same service (45). Where water is scarce and expensive, consideration of the availability and cost of the makeup water, which would be

needed for plant operation with evaporative cooling, can lead to economic advantages with the employment of non-evaporative devices. When this is the case, both turbine and condenser design must be modified to operate most efficiently with water to air heat exchanger cooling of the circulating water (46). There are also hybrid systems employing both evaporative and non-evaporative devices in the same system with various modes of operation. These systems, sometimes referred to as wet/dry systems, are designed to reduce makeup water requirements while exhibiting the increased cooling characteristic of the evaporative configuration (47, 48).

Since water for makeup is readily available at the MIT site at a moderate cost it was decided to pursue the design of an evaporative cooling system for waste heat dissipation. When the design parameters and costs for this system have been determined, its estimated makeup requirements will be used to substantiate this decision.

Evaporative cooling devices may further be subdivided into cooling ponds and cooling towers. Cooling ponds employ once-through cooling with a man-made reservoir. Heat dissipation from the pond may be enhanced by the incorporation of water spray over and into the pond. The heat dissipation capacity of the pond is a function of the pond's surface area, atmospheric conditions, and the characteristics of the spray if it is employed. The land area required is usually the limiting factor, especially where urban sites are concerned. Area requirements range roughly from 1 acre with no spray to

0.05 acres with spray, per MW of generating capacity (49). For use with the smaller extraction steam turbine plant at MIT a cooling pond with spray would have to be about an acre in area. Land is simply not available for this use. The obvious alternative is the use of cooling towers.

Cooling towers are available in a multitude of configurations. The choice of a particular configuration is keyed to the service required, including flow rates and temperatures, the local weather conditions, the physical arrangement and conditions at the site, and the environmental and safety regulations in the area. Gurney and Cotter (50) give some general characteristics which can be used for qualitative evaluation of a particular configuration. For application at MIT a mechanical draft cooling tower in which air flow is provided by a fan was selected for the conception design based on the following:

1. Available land area. The area in the vicinity of the proposed location for the MIT power plant is an obstructed urban area. Both atmospheric towers, where air movement in the tower depends upon wind, and natural draft towers, where air movement is dependent upon the stack effect of a chimney with the less dense warm air rising within the tower, require large unobstructed areas relative to the requirements of the mechanical draft tower.
2. Required service. With a condensing application, lower temperature cooling water will result in more

efficient plant operation. For similar (physical) sized towers and identical atmospheric conditions mechanical draft towers will yield a lower temperature circulating water. By controlling fan speed, close regulation of outlet water temperature is possible with a mechanical draft tower. This control is difficult with atmospheric and natural draft towers.

3. Capital costs. The tall chimney configuration of the natural draft tower makes its construction more expensive than mechanical draft towers. Atmospheric towers are also tall but are less expensive to build than natural draft towers and have capital costs comparable to those of mechanical draft towers.
4. Site configuration and operating experience. Experience has been entirely favorable with the mechanical draft towers now operating at MIT. There is a planned site for construction of additional mechanical draft towers adjacent to the power plant location.

With the comparable design and the selection of components for the proposed towers similar to those which were employed in the existing mechanical draft towers, the single tower operating characteristics will be much the same as those experienced and found acceptable with the present installation. There are, however, additional considerations which are necessary with multiple tower arrangements, arising

from the combined effects of more than one tower. These include vibration and fan noise, recirculation (where the air out of a tower subsequently goes into the inlet of the same or an adjacent tower), and plume behavior. A plume is the cloud-like exhaust air stream released from an operating tower. In some configurations plumes from adjacent cooling towers have been found to reinforce one another, resulting in a higher but more dense exhaust stream or cloud (51). The final design should investigate these aspects of siting in more detail than the conceptual design presented below.

3.3 Specification of Mechanical Draft Towers

Prior to determining the specification for optimal mechanical draft cooling towers in a specific application, it is helpful to review cooling tower terminology and heat transfer theory, which are somewhat unusual. With the interrelationships between the heat transfer variables defined, optimization of the heat sink parameters will determine the cooling tower's design performance specifications and indicate appropriate models for predicting off-design performance.

3.3.1 Mechanical Draft Cooling Tower Terminology and Theory

Specific configurations for mechanical draft cooling towers are generally defined by one out of four possible combinations characterized by fan location -- at the air inlet or outlet, and by relative direction of air

and water flow, crossflow or counterflow. These four configurations are illustrated in Fig. 3.5. They are:

1. Forced-draft crossflow tower. Fans are situated at the air intake and blow ambient air into the tower and across the falling water. In this configuration, the rotating mechanical equipment is near the ground on a firm foundation with less vibration and in a comparatively dry air stream. There is a tendency for ice to form on the fan blades during the winter. This type of tower can present difficulties with recirculation at low outlet air velocities since the low pressure fan inlet is accessible . Crossflow towers are typically shorter but with a larger cross-sectional area than comparable counterflow towers.
2. Forced-draft counterflow tower. It is similar to the forced-draft crossflow tower except that air flow is directed upward such that most of the heat transfer occurs with water and air in counterflow. Counterflow towers present the most efficient heat transfer. Advantages and disadvantages due to fan location and construction are similar to those of the forced-draft crossflow tower above.
3. Induced-draft crossflow tower. Fans are situated at the air outlet, usually on top of the tower, drawing air form the inlets across the falling water and then up to the outlet. Induced-draft configurations are more likely to present vibration and maintenance

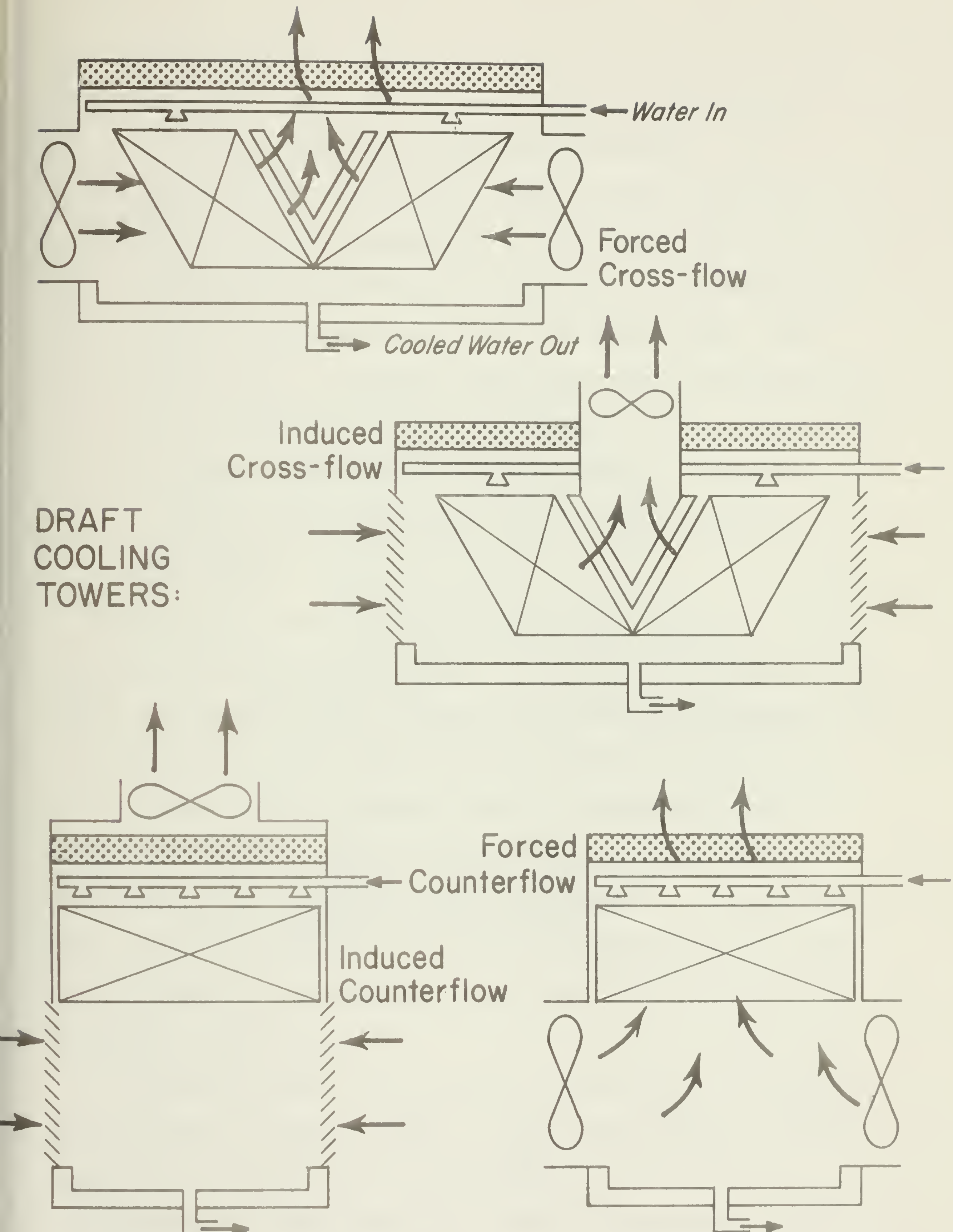

Figure 3.5. Major configuration for mechanical draft cooling towers.

problems since the fans are located high on the tower in a hot, humid air stream. The high air outlet velocities make recirculation improbable.

4. Induced-draft counterflow tower. Fans create vertical air movement in opposition to the water flow. This is the maximum performance arrangement with the coldest water contacting the coolest and driest air. Less ground area is required than with other configurations since there are no fans on the side and the major heat transfer area is above the air inlets. Mechanical components are not always easily accessible for maintenance.

The induced-draft counterflow cooling tower was chosen for use in the conceptual design and feasibility study at MIT since it represents the optimum heat transfer arrangement and is compatible in size with the area available adjacent to the existing towers, numbers 3 and 4. Experience with the existing induced-flow towers has presented no exceptional difficulty with vibration, noise, maintenance, or recirculation. With few changes, the design specifications for this type of tower are determined in the same manner as for any mechanical draft tower.

A detailed sketch of a typical induced-draft counterflow cooling tower is shown in Fig. 3.6 for the purpose of locating important components. The tower shown is a 2-cell configuration in which water flow and fan speed may be controlled separately for each cell. The basin and return

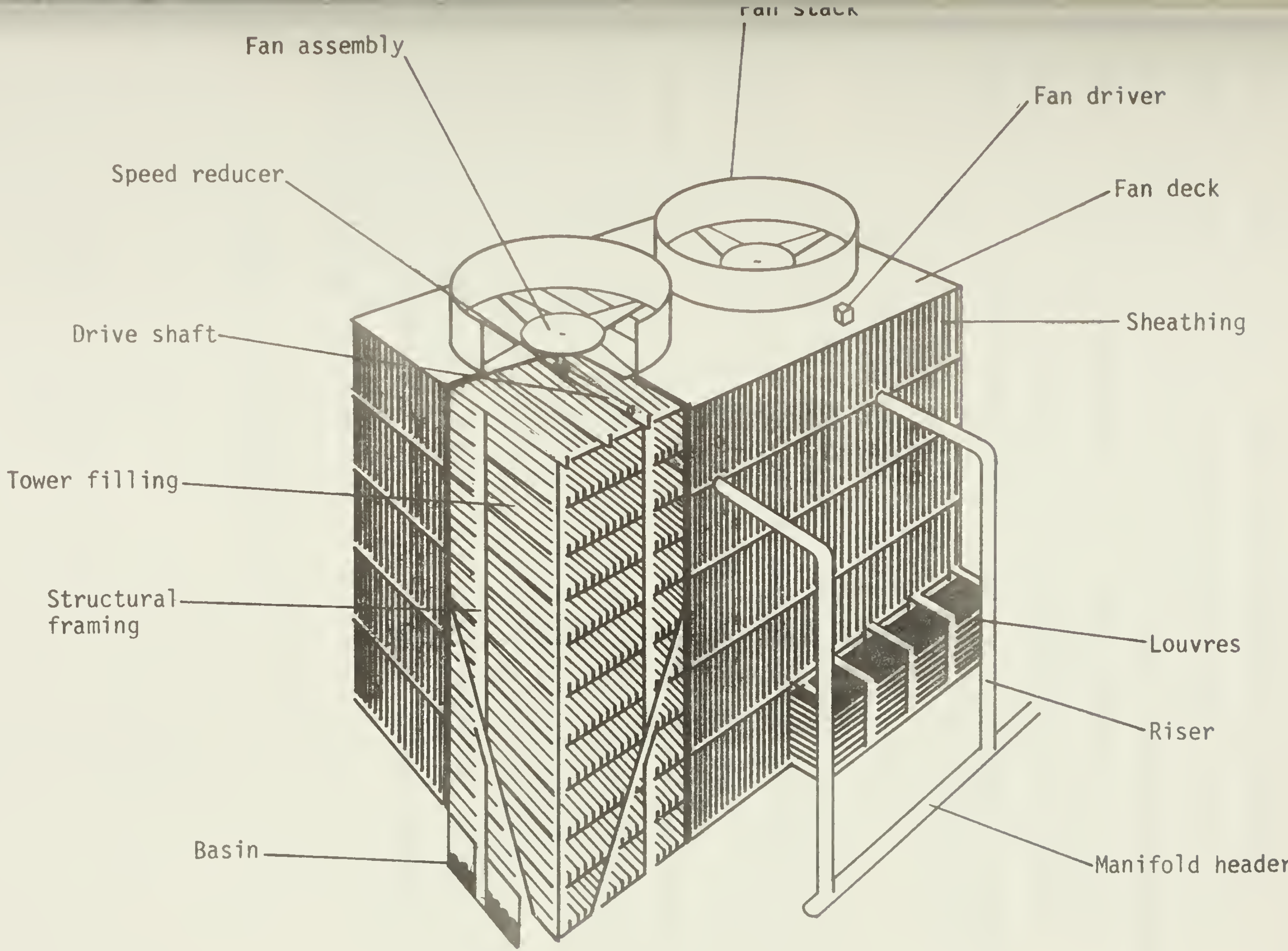

Figure 3.6. Typical induced draft counter flow cooling tower detailing major components.

piping is common to both cells. The major portion of the heat transfer from the water to air takes place in an area of the tower in which there are specially constructed physical obstructions to water flow. These obstructions are designed to break up the water flow into small droplets and are called packing or fill. The particular packing configuration and material of construction are dominant among the variables used in cooling tower design to establish a given heat transfer capability (52). The nomenclature of the remaining components is self-explanatory.

The interrelationships between the variables which must be specified for cooling tower selection or design are best understood by following the development of the most widely accepted cooling tower heat transfer theory, largely due to Merkel (53). This theory has several key simplifications but has been shown to produce results which predict tower performance consistent with the ability to establish uniform conditions (such as constant air and water flow rates) and experimental uncertainties (54). The results of Merkel's development may be stated as:

$$KaV/W = \int_{t_2}^{t_1} [dt(h' - h)]$$

or

$$KaV/G = \int_{h_2}^{h_1} [dh/(h' - h)]$$

where: Ka = product of mass transference coefficient and water droplet surface area per unit packing volume

V = volume of the packing or fill in tower

W = inlet water flow rate

G = inlet air flow rate

t_1 = water inlet temperature

t_2 = water outlet temperature

h' = enthalpy of saturated air at the water temperature

h = enthalpy of moist air

h_1 = enthalpy of saturated air at tower air inlet temperature

h_2 = enthalpy of saturated air at tower air outlet temperature

The important relationship for the system designer to glean from these equations is that the driving force for heat transfer from water to air in a wet cooling tower is the enthalpy difference between saturated air at the water temperature and saturated air at the air temperature. Thus, the water cannot be cooled below the inlet saturated air or wet bulb temperature. The difference in temperature between the water out of the tower and the air inlet wet bulb temperature is called the approach; for 75°F water outlet temperature and 60°F wet bulb air inlet temperature, the approach is 15 F°. Reference to a psychrometric chart will indicate that there is a higher enthalpy difference for a given approach at higher wet bulb temperatures since there is a greater amount of water vapor in saturated air at

higher temperatures. Closer approaches are possible at higher wet bulb temperatures and with all other variables constant, an increase in ambient wet bulb temperature will cause water temperature out of the tower to increase, but with an increase less than the rise in wet bulb temperature.

The 'KaV' term in Merkel's equation along with the air ('G'), and possibly water ('W'), flow rates are the cooling tower design variables. 'KaV' is analogous to the heat transfer coefficient in conduction heat transfer with flowing fluids and is determined by the packing configuration and flow rates at design conditions. This term is the source of difficulty in accurately predicting off-design performance for a given cooling tower in that it includes effects from water air flow interactions as well as from the water and air temperature profiles within the tower. Accepted practice is, then, to specify worst case or limiting conditions as the design specifications. These must include the highest ambient wet bulb temperature in which the tower is to operate, the design maximum water flow rate, and the highest tower water outlet temperature which can be tolerated for design operation. Conformance to these specifications will produce a cooling tower able to dissipate full-load waste heat at the design wet bulb temperature. An important issue is to determine the peak loading on the tower and whether this may or may not occur at the time of the maximum ambient wet bulb. System performance should not be penalized when full duty is required, but excess waste heat dissipation capacity based

on design conditions which will not exist in actual operation must be avoided.

With the co-operation of cooling tower vendors a semi-empirical system of cooling tower design and evaluation parameters was developed and published by Kelly (55). This system, which will be used below, is based on the "Tower Unit", TU, which is numerically equal to the product of water flow rate in gallons per minute and a "Rating Factor", RF, which is obtained from Rating Factor curves. The "Rating Factor" is a measure of the degree of difficulty in obtaining a given tower water inlet minus outlet temperature difference, called the range, with a specified approach at a particular wet bulb temperature. Thus, the "Rating Factor" is a relative indication of the KaV/W value required by Merkel's equation for specific conditions with the number of tower units corresponding to a relative or normalized 'KaV'. Experience has shown that cooling tower size, cost, and fan horsepower requirements each relate proportionally to the number of tower units. The large amount of base data and demonstrated accuracy make this system a valuable aid to the designer in specifying cooling tower requirements.

3.3.2 Optimization of Cooling Tower Design Parameters for Application with Extraction Steam Turbine Plant

The waste heat dissipation system for use with the extraction steam turbine plant includes, as a major component, not only the cooling tower but also the steam condenser.

Clearly, considering initial costs, an expensive cooling tower producing low cooling water temperatures will require less heat transfer surface and thus a less costly condenser than a less expensive cooling tower. This simple cost trade-off is the basis of the design parameter optimization presented here. The other major factors to be considered are pump and piping initial costs and the cost of the power directly required to operate the waste heat disposal system. This evaluation produces optimal design parameters for a cooling water system to support full load design operation of the steam extraction turbine. The resulting design parameters may not be the optimum for year-round operation of the power plant since off-design operation of the cooling water system and its effect on power plant efficiency are not considered in the optimization. A more complete optimization including the turbine back-pressure effect for coincident load and weather conditions can be conducted with the same computer routine which is to be used for the feasibility study if the initial study indicates a potential advantage for the extraction steam turbine plant.

The design maximum exhaust pressure for the extraction steam turbine is specified as 3 inches of mercury corresponding to a condensing temperature of 115°F. Assuming a minimum 5 F° terminal temperature difference between the condensate and the circulating or cooling water in the condenser, the maximum design cooling tower inlet water temperature is 110°F. The assumed 5 F° terminal temperature difference is the

minimum recommended by the Heat Exchange Institute (56). These conditions specify the heat source and cooling water inlet temperature for the cooling system design.

The value of the ultimate heat sink for cooling systems employing cooling towers has been shown to depend upon the ambient wet bulb temperature. Meteorological data analyzed to establish design conditions for space comfort indicate that in the Cambridge, Massachusetts area a wet bulb temperature of 76°F should be exceeded for less than 1.0% of the hours in a year (57). This 1% criterion is the design wet bulb condition usually specified for power plants with all but the most vital service requirements. This is, then, the reasonable choice for the MIT installation, especially since the load profiles indicate that peak electrical demand occurs during the afternoons in summer months, which could be coincident with the highest wet bulb temperatures. Considering the capability of the Facilities Management System to reduce non-essential electrical load, the capacity to supply the design for a guaranteed 99% of the time is considered acceptable. Thus, 76°F is selected as the design wet bulb temperature for full load operation.

Within the established temperature bounds a few reasonable assumptions will establish a unique set of cooling system design parameters for each cooling tower choice as specified by either the range or approach. Fig. 3.7 depicts the temperature relationships for cooling system heat transfer with several different values of cooling tower

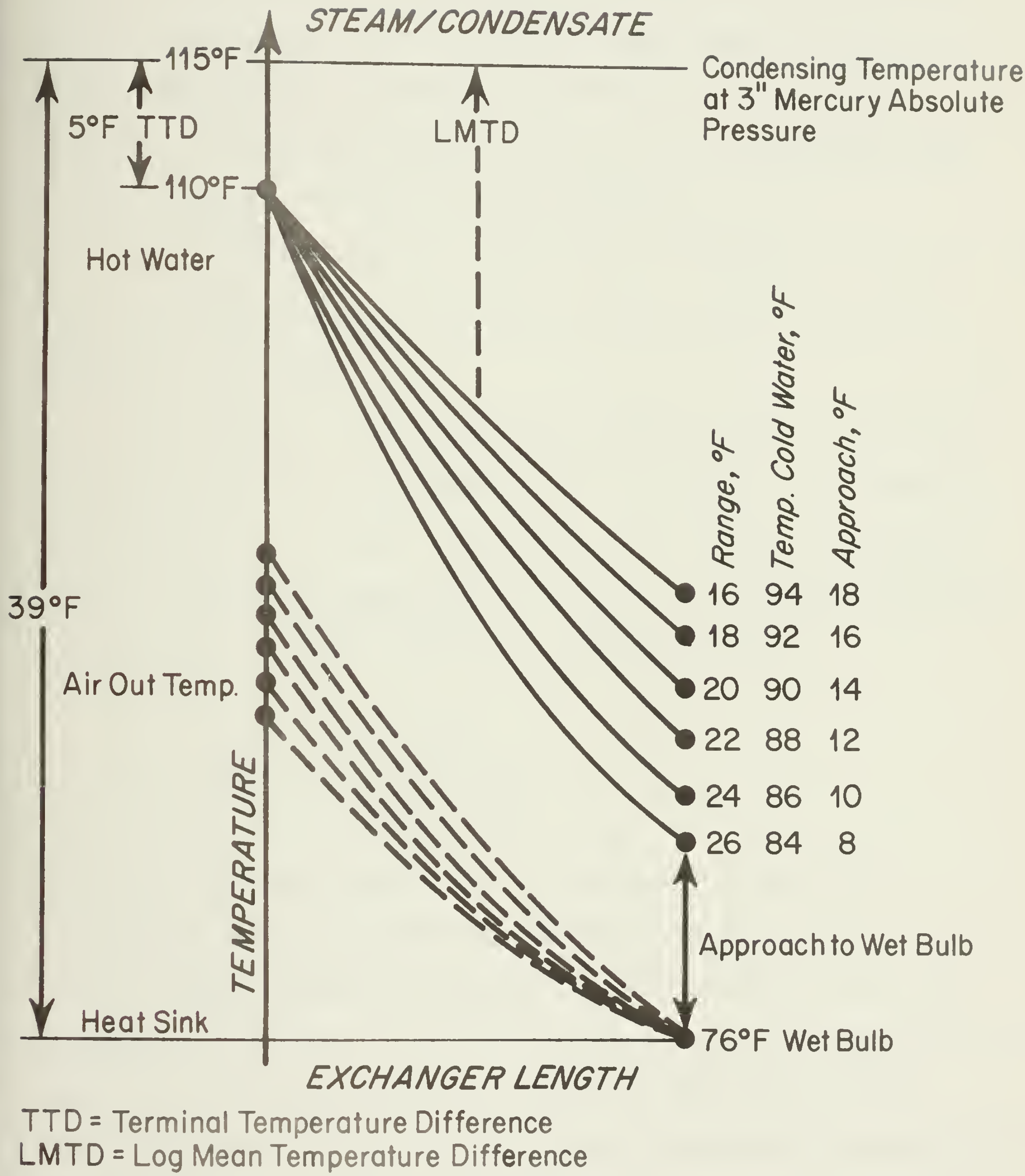

Figure 3.7. Steam condenser-cooling tower heat transfer diagram.

range and approach. An 8 F° approach is the minimum design value recommended by the cooling tower vendors. Neglecting any heat transfer in the connecting piping and considering steady state operation the cooling tower range is equal to the cooling water temperature difference across the condenser. Cooling water flow rate, W, is determined from a simple energy balance on the condenser:

$$\text{Rejected Heat} = (W) \times (C_p) \times (\text{Range}),$$

where the specific heat of the cooling water, C_p, is assumed to be unity.

The sizing of the condenser is based on the log mean temperature difference, LMTD, and the overall heat transfer coefficient, U_o, in the condenser. The LMTD is determined by the condensing temperature T_c, and the condenser cooling water inlet, T_1, and outlet, T_2, temperatures by:

$$\text{LMTD} = \frac{T_1 - T_2}{\ln\left(\dfrac{T_c - T_1}{T_c - T_2}\right)},$$

where T_1 is established from the terminal temperature difference and the condensing temperature and then T_2 is set by the range. From the design recommendations of the Heat Exchange Institute (56), an average value of 650 BTU/hr-ft^2-°F LMTD may be assumed for U_o. This is a mid-range design value for the tube type steam surface condensers corresponding to about 6 ft/sec cooling water velocity in the condenser tubes. The required condenser heat transfer area, A, is then

determined from the relation;

$$\text{Rejected Heat Rate} = (U_o) \times (A) \times (\text{LMTD})$$

This condenser heat transfer area is an accepted basis for estimating the initial cost of condensers with a 1977 value of approximately $6/sq.ft.

Cooling tower costs are estimated on the basis of the required tower units. The cooling tower "Rating Factor", RF, is established for a particular wet bulb temperature by the range and approach with the curves from Kelly (55) shown in Fig. 3.8. The number of required "Tower Units", TU, is, then, simply the product of the "Rating Factor" and cooling water flow rate, W. The incremental cost per tower unit, covering the labor and meterials for erection of a wooden (Douglas fir) cooling tower including fans, fan motors, and piping within the tower but excluding the cost of the coldwater basin, wiring, and piping external to the tower was approximated by Kelly at $8/TU in 1975 (55) and revised to $10/TU in 1977 (58). These estimates were obtained from the averaging of costs for a large number of wooden cooling tower installations of various sizes and in different locations.

The cooling tower cost for the MIT installation is expected to deviate from this average due to three factors. Labor and transportation costs for the MIT case will no doubt be higher than the national average because of the prevailing wage scales in the Cambridge area and the distance of the site from a probably mid-US shipping point. The most

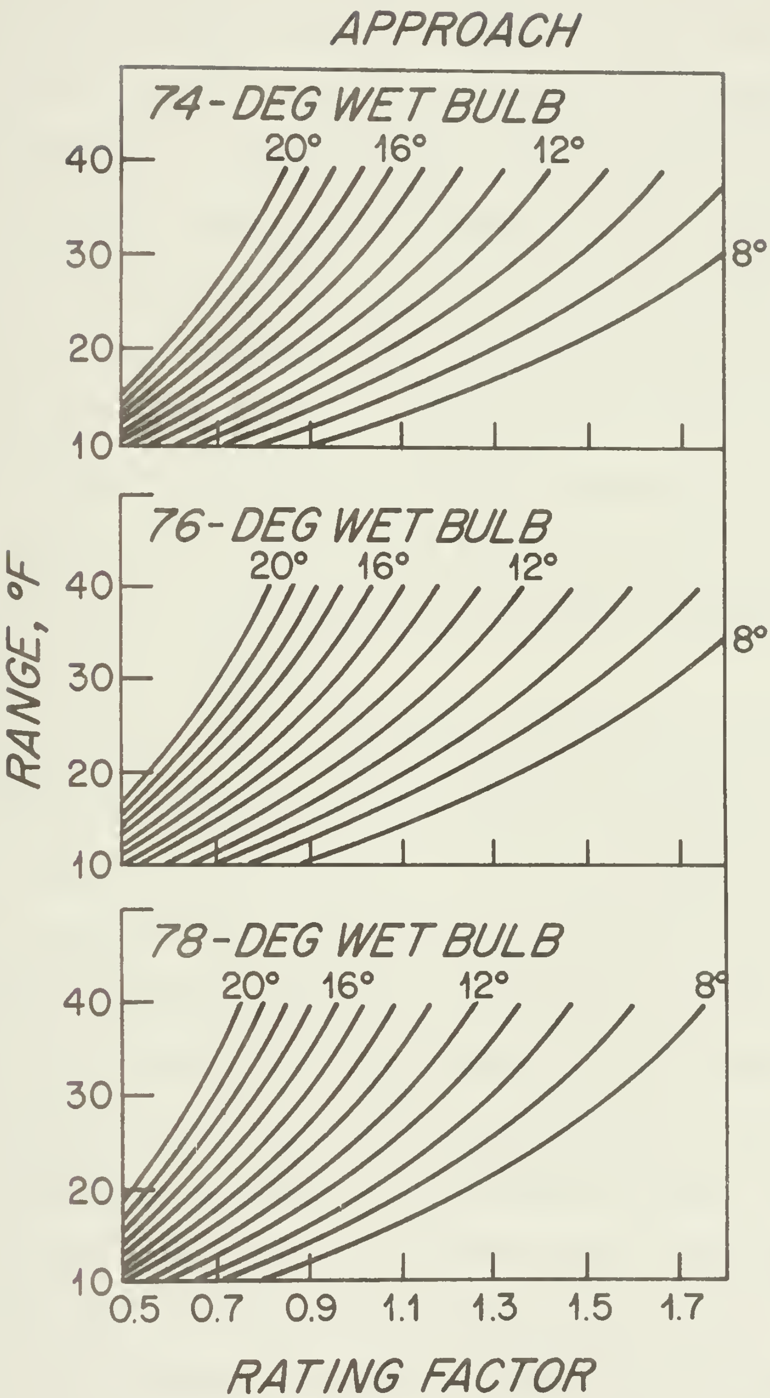

Figure 3.8. Rating factor curves.

signigicant factor, however, will be the materials selected for the cooling tower construction. Comparative experience at MIT with both wooden towers, towers 1 and 2, and steel towers, towers 3 and 4, had led to a decided preference for steel construction. Additionally, the proposed site for additional towers, adjacent to towers 3 and 4, will require cooling tower designs with compatible, compact physical dimensions. These compact tower designs are best achieved through the use of packing or fill material which promotes more efficient heat transfer than the conventional wood packing. Thus, selection of the more expensive steel tower construction with non-wooden fill material similar to towers 3 and 4 is indicated.

With contract cost information available for the previous cooling tower construction at MIT (59, 60), it was decided to base cost estimates on an incremental cost per tower unit derived from the cost of cooling tower units 3 and 4. Calculations, detailed in Appendix F, based on the specified 32 F° range, 9 F° approach, and 76°F ambient wet bulb temperature for both towers, show tower 3 with 7000 gpm water flow to be a design of 10,990 Tower Units. Tower 4 with 7950 gpm water flow is a 12,482 Tower Unit design. Initial tower costs exclusive of basin, wiring, and piping external to the towers were $175,777 for tower 3, quoted in June 1971 and $234,376 for tower 4, quoted in October 1973. These costs were brought up to 1977 dollars using the same 6% average annual escalation indicated by Kelley (58). On this

basis incremental cost for tower 3 would be projected at \$22.69/TU if contracted in 1977, and tower 4 would be \$23.71/TU. For the purposes of capital cost estimation for the proposed tower installation a conservative incremental base cost of \$24/TU is used. Similarly, the cold water basin sizing and cost factors were revised from Kelley's recommended 0.123 ft^2/TU and \$10/ft^2 to 0.141 ft^2/TU and \$15.52/ft^2 from the experience with basin construction for tower units 3 and 4.

The materials and installation costs for the pumps and piping system exclusive of the cooling tower and condenser are based on the cost of the similar portion of the installation of cooling tower 4. The detailed cost of the pumps and piping associated with the installation of cooling tower 4 were obtained from contract records (61) with the appropriate costs shown in Appendix F arrived at in consultation with Mr. R.F. McKay of the MIT Department of Physical Plant. The pump and pipe system cost for the proposed installation was then computed by considering the cost to consist of two parts; the fixed portion, \$277,000 in 1977, containing the cost of all tasks which would not be affected by system sizing and cooling water flow rate, and the variable portion, \$281,000 for 7950 gpm flow rate, which was applied in direct proportion to system flow rate when compared to the 7950 gpm design in cooling tower 4.

Cooling system yearly operating costs were estimated using approximate pump and fan horsepower requirements, periods of operation, and an assumed power cost. The cost of power for operation of the cooling system was assumed to be \$0.0269/hp-hr, corresponding to \$0.036/KW-hr which is the 1977

average cost of commercial electric power at MIT. This cost is not meant to be the actual cooling system operating cost since the power for the installation will be provided by the total energy plant and its cost assessed on the basis of total plant costs; however the operating costs computed on this basis will be indicative of the relative costs for the variation of cooling system design parameters.

The required fan horsepower was estimated based on the tower unit value of 0.016 hp/TU since the fans and associated drive gear reducers for this tower incorporated several design advances over those in tower 3 and are more indicative of applications in present cooling tower design. Fan power cost, then, is given by:

Fan Power Cost ($/yr) =
(.0269/hp-hr) x (fan hp) x (fan utilization, hr/yr)

Fan utilization must reflect the fact that the fans are variable, two-speed units and will be operated at full speed only part of the time; in fact during cold weather they may not be operated at all. To account for this, fan utilization is assumed for one half the hours of plant operation or 4380 hrs/yr. This assumption generally compares well with the operating history of the existing towers. Actual fan operating times will be highly dependent upon fan control settings and weather conditions.

The required cooling water pumping head must be sufficient to overcome the head lost in the cooling tower as well as the frictional drag in the remainder of the cooling

water flow path. The design head requirement for tower 4 is 42.0 feet of water at the plane of the bottom of the cold water basin. Since the piping system cost estimates are based on constant fluid velocity it is consistent to consider the head loss due to fluid friction constant even though this quantity is highly dependent upon the specific piping layout and final plant configuration. The present cooling water system including tower 4 has about 25 feet of water head loss when operating alone at 7950 gallons per minute flow. A portion of this piping system is sized to accommodate almost twice this flow to include tower 3 operation; the head loss would be somewhat higher were this portion of piping sized for 7950 gpm. From these base factors it was decided to assume a required 75 feet of water pumping head at rated flow for the initial cooling system optimization. The pump operating costs are then determined from the horsepower required to provide a given water flow rate, W, at the required head, 75 feet, from:

$$\text{Horsepower required (hp)} = \frac{W \frac{\text{gal}}{\text{min}} \times 8.33 \frac{\text{lb}}{\text{gal}} \times 75 \text{ ft}}{33000 \frac{\text{ft-lb/min}}{\text{hp}}}$$

Assuming a pump efficiency of 85% and a motor efficiency of 92% pumping costs, \$/year, are given by:

$$\text{Pumping Cost (\$/yr)} =$$

$$\frac{\text{hp required} \times \text{utilization} \left(\frac{\text{hr}}{\text{yr}}\right)}{.85 \times .92} \times \text{Power cost} \left(\frac{\$}{\text{hp-hr}}\right)$$

Pump utilization is assumed to be 8760 hours per year corresponding to continuous plant operation. More detailed evaluation of cooling system operational costs can be performed in conjunction with simulation of plant operation; however, the simple evaluation presented here will be sufficient to determine design parameters for the initial feasibility study.

On the basis of these assumptions and estimations cooling system costs are calculated for each of the extraction steam turbine plant capacities being considered, the 10 MW(e) unit with a maximum rejected heat rate of 28.29 MW(t) and the 15 MW(e) unit with a maximum rejected heat rate of 41.61 MW(t), over a span of possible design parameters characterized by various values of the approach. These calculations are summarized in the optimization curves shown in Figs. 3.9 and 3.10 where the net present worth or cost of the cooling system, calculated assuming a 30 year cooling system lifetime and a 6% annual interest rate, was selected for final cost comparison. Detailed results, tabulated in Appendix G, include yearly cost data determined with the same assumptions. For both cooling systems selection of an approach between 9 and 10 F° is indicated. The lower 9 F° approach is selected as being nearest minimum cost. This gives a design range of 25 F°. This is then to be considered as the base design for each application with a cooling water flow rate of 7726 gpm for the rated 10 MW(e) installation, and 11364 gpm for the 15 MW(e) case.

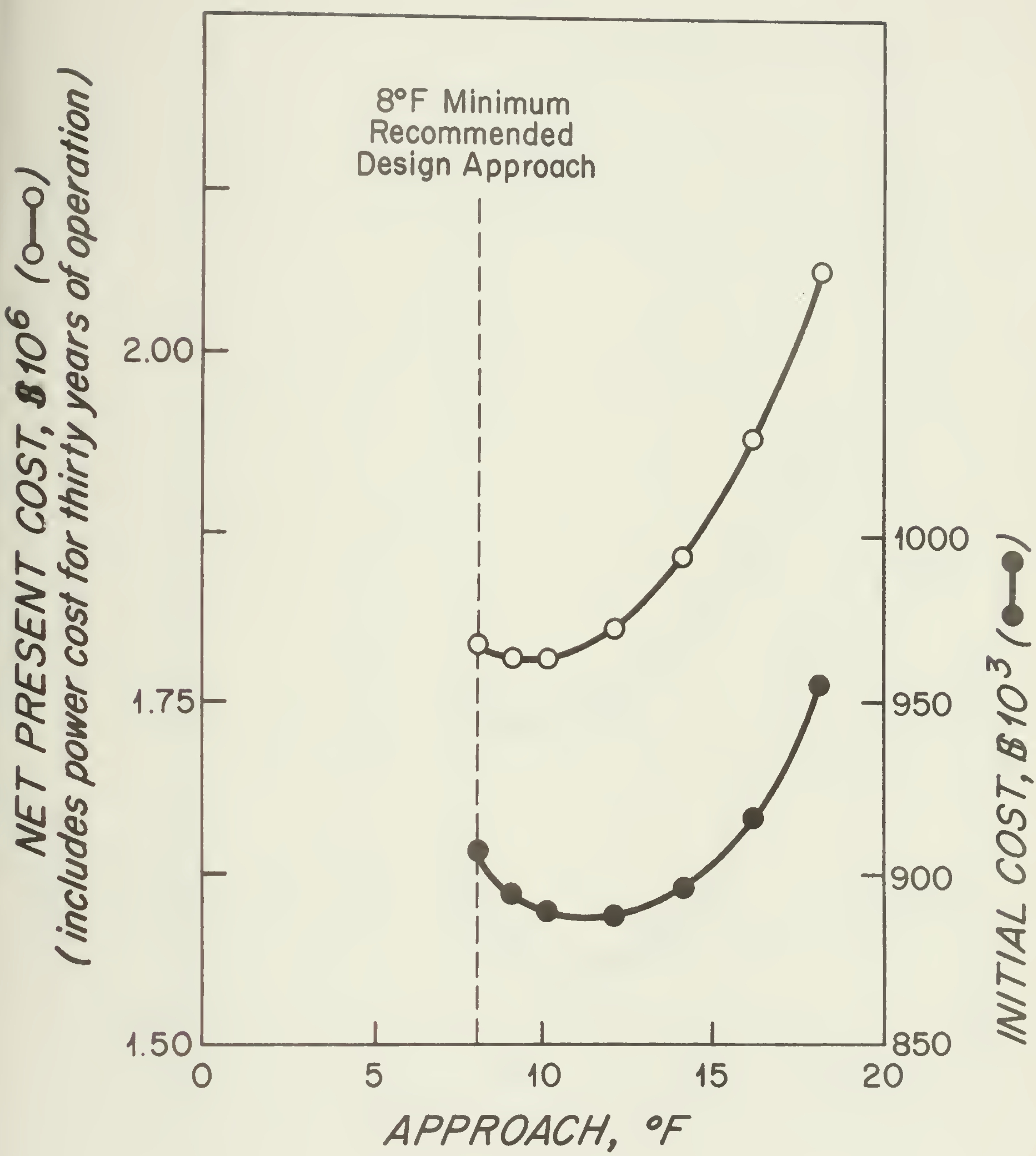

Figure 3.9. Cooling system optimization curve for 10 MW(e) extraction steam turbine installation.

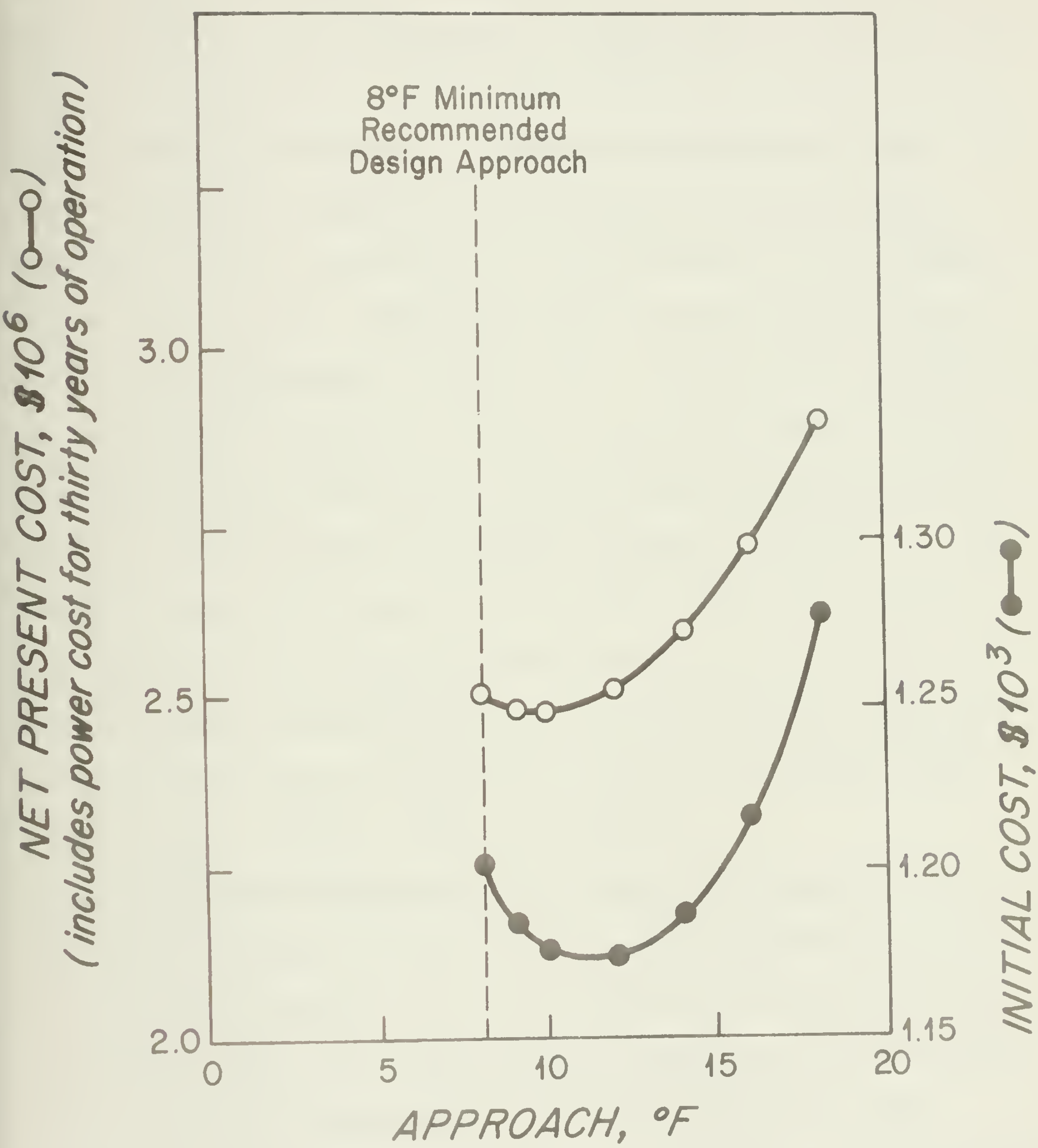

Figure 3.10. Cooling system optimization curve for 15 MW(e) extraction steam turbine installation.

The optimization calculations are based on several factors which require a degree of judgement for the estimation of their values. It is important to establish the effect of a possible variation of each of the estimated factors on the optimum design parameters. This was accomplished simply by reperforming the optimization calculations varying each estimated factor separately to the extremes of its probable value. The results, which were the same for both installations, are summarized in Table 3.2. Clearly the optimization is relatively insensitive to the variation of any one factor determining the initial cost of the cooling system, with the optimal system having an approach between 9 and 10 F° . The shift in the optimum approach to 10 F° with with extreme variations of the required pumping head, cost of power, and value of money characterized by the interest rate points out areas for additional analysis in a final extraction steam turbine plant design. These areas are;

1. Accurate determination of fuel cost penalty attributable to the cooling system operation. This should include the cost of the fuel to generate the electric power required for operation of the cooling system in comparison with the differential fuel consumption due to the variation of turbine efficiency with condenser pressure. This determination can be made with simulations including coincident weather and power demand conditions for different sets of cooling system parameters.

TABLE 3.2

EFFECT OF THE VARIATION OF BASIC QUANTITIES ON THE OPTIMIZATION OF COOLING SYSTEM DESIGN PARAMETERS

Factor	Base Value	Variation	Optimum Design Approach
Incremental Tower Cost	\$24/TU	+25% - \$30/TU -58% - \$10/TU	10 F° 9 F°
Cold Water Basin Cost	\$15/52/ft^2	+29% - \$20/ft^2 -35% - \$10/ft^2	10 F° 9 F°
Condenser Overall Heat Transfer Coefficient U_o	650 BTU/hr-ft-°F	+15% - 750 $\frac{\text{BTU}}{\text{hr-ft-°F}}$ -31% - 450 $\frac{\text{BTU}}{\text{hr-ft-°F}}$	10 F° 9 F°
Condenser Incremental Cost	\$6/ft^2	+33% - \$8/ft^2 -33% - \$4/ft^2	9 F° 10 F°
Pump and Piping Cost	\$277,000 + $\frac{\$35.35}{\text{gpm}}$(flow)	+50% -50%	9 F° 10 F°
Incremental Fan Horsepower	.016 hp/TU	+25% - .02 hp/TU -25% - .012 hp/TU	10 F° 9 F°
Pump Head	75	+33% - 100 ft -33% - 50 ft	9 F° 10 F°
Power Cost	$\frac{\$.0269}{\text{hp-hr}} = \frac{\$.036}{\text{kw-hr}}$	+50% - $\frac{\$.0404}{\text{hp-hr}} = \frac{\$.054}{\text{kw-hr}}$ -50% - $\frac{\$.0135}{\text{hp-hr}}$ $\frac{\$.018}{\text{kw-hr}}$	9 F° 10 F°
Interest Rate	6%	8%/15%/20%	9 F°/10 F°/10 F°

2. Accurate determination of the required cooling system pumping requirements obtained from final system piping configurations. This determination will ultimately involve a complex tradeoff between cooling system fuel penalty and piping system initial cost.
3. Realistic consideration of the method of financing the project to determine an appropriate interest rate for use in evaluating the relative worth of initial costs and future operating expenses,

With this additional resolution the cooling system design parameters can be refined; however, it is expected that the final parameters will be very nearly those determined above for the feasibility analysis.

To substantiate the decision to use an evaporative cooling tower, the annual cost for makeup water is computed. With a water cost of $0.36 per 100 cubic feet, in Cambridge, and a conservatively high makeup rate of 3% (MIT's average makeup for operation of towers 3 and 4 has been about 1.7% but they are not subject to continuous operation), approximately $59,000 and $86,000 would be spent for makeup cooling water yearly with the 10 and 15 MW(e) plants respectively. Considering this as the annual savings with a dry tower, $807,000 and $1,187,000 additional capital is available to cover the added cost of the dry tower installation assuming a 6% "value of money" interest rate. In both cases this allows for a dry cooling tower costing under four times the cost of the proposed wet tower and basin. With

the conservative assumptions in this calculation and the indications that the cost of dry cooling tower systems for generation facilities of about 10 MW(e) tends toward the high end of the range of 3 to 5 times the cost of evaporative towers, the initial decision to opt for a wet tower appears to be a valid one for the feasibility analysis.

3.3.3 Conceptual Design of Cooling Water Facilities for Use with the Gas Turbine Plant

Even with the very approximate design data for the gas turbine installation it is apparent that the requirements for cooling water facilities to support operation of this plant are minimal compared to either the extraction steam turbine or dual-fuel engine configurations. With the small magnitude of cooling water flow required the situation is ideal for incorporating oil cooling by heating of makeup feedwater to the waste heat boiler. The MIT steam distribution system requires an average of about 8% makeup. With the 1976 minimum steam flow of 38,000 lbm/hr minimum makeup feed requirements are approximately 76 gpm which exceeds the maximum oil cooler requirements of 12 gpm. Ample cooling can be provided for the gas turbine installation by the simple incorporation of preheating makeup feed with no storage facilities required.

3.3.4 Specification of a Cooling Tower for Use with the Dual-Fuel Engine Plant

The dual-fuel engines are supplied by the vendor equipped with a cooling water pump and appropriate heat exchangers; the only major component required to complete the cooling system is the heat sink or cooling tower. Cooling water requirements are specified as 1200 gpm flow with a maximum temperature at the supply to the engine of 85°F. The design maximum rejected heat rate of 4.75 MW(t) would result in a minimum cooling water temperature out of the engine of

112°F. The cooling tower specifications are, then, an approach of 9 F° at a 76°F design wet bulb temperature, with a 27 F° range and 1200 gpm flow rate. With the 76°F wet bulb curve of Fig. 3.8 it is seen that this tower would have a "Rating Factor" of 1.45 and thus would be a tower of 1740 "Tower Units". This is a measure of the cooling tower waste heat dissipation capability required for each engine. The tower units required to service the multiple engine installation increase in direct proportion to the number of engines employed in the facility.

Two cooling tower configurations were investigated for both the three and four engine installations. The first configuration, similar to that proposed for the extraction steam turbine installation, would consist of a multiple cell, induced-draft, counterflow cooling tower with each cell designed to provide cooling for one engine. The other configuration was suggested from the cost advantage shown for prefabricated, packaged cooling towers in small capacity applications (45). These towers, though termed 'double-flow' by one vendor (62), are essentially single cell, induced-draft, crossflow cooling towers which are factory assembled and then simply hoisted into position and bolted down at the site.

These possible configurations were compared on the basis of the cost of materials, transportation, construction, and erection of the tower and cold water basin (for pre-fabricated towers this is one unit). The cost comparison did not include

projected operating costs or the cost of wiring, controls, and piping connections to the engines since it was judged that these costs would be approximately the same for both tower configurations at the same roof top site and servicing the same number of engines.

Since the data used for estimating the cooling tower cost for application with the extraction steam turbine was based on the construction of larger capacity towers than those which would be required for the dual-fuel engine installation, new and approximate data was obtained from Mr. G.M. Kelly of the Marley Company (63) for the counterflow tower constructed on site. Kelly indicated that fireproof towers in three or four-cell configurations would cost about $15 or $16 per tower unit erected on the site, where the higher incremental cost would apply to the four-cell tower. He also stated that his original (55) incremental basin area requirement of 0.123 square feet per tower unit should still apply to this smaller tower. Basin cost was then projected on the basis of experience with towers 3 and 4 at $15.52 per square foot. With this data the estimated cost of a single tower to provide cooling for either a three or four engine installation was determined. This cost is shown in Table 3.3, along with the estimated cost of 3 and 4 single, packaged towers. The packaged tower used as the basis for the cost and dimensional data is the Marley Cooling Tower Model 8616 (65).

The total cost figures shown in the table dramatically

TABLE 3.3

FIRST COST COMPARISON OF COOLING TOWER OPTIONS FOR USE WITH DUAL-FUEL ENGINE TOTAL ENERGY PLANT

Single Multiple Cell Cooling Tower Constructed and Erected on Site:

	3-Engine Installation	4-Engine Installation
Tower Units	5220	6960
Tower Cost, $	$78,300	$111,360
Basin Area, ft^2	642	856
Basin Cost, $	$9,964	13,286
Total Cost, $	$88,264	$124,646

Proposed Tower Installed on Site:

Total Cost at $18,000/unit	$54,000	$72,000
Basin Area at 131 sq.ft/unit	393	524
Roof Area at 197 sq.ft/unit	591	788

favor selection of the multiple packaged tower option. The cost which was estimated on an incremental basis per "Tower Unit" for the single multicell tower has a greater potential for error than the approximate cost quotations used for the packaged units; however, the total cost difference represents 39% and 42% of the multicell tower cost for the 3 and 4 engine installations respectively. This is too large a difference to be attributed to an error in the incremental cost calculations. Clearly the dual-fuel engine plant configurations should include the use of individual packaged cooling towers.

3.4 Performance Models of Waste Heat Dissipation Systems

Cooling tower performance will vary throughout the year primarily as a function of the ambient wet bulb temperature. The parameters which will change and be reflected in the over-all plant performance are the cooling water temperature and the fan speed. With a control system postulated to sense cooling water temperature and control the fan speed and flow through the tower, the variable parameters are related. The cooling water temperature is reflected in plant performance for the case of the extraction steam turbine by establishing the condenser pressure and thus the electrical generation efficiency of the turbine-generator. Cooling of the dual-fuel engine has a negligible effect on its generation efficiency so long as temperatures are maintained within the allowed band to prevent excessive wear or failure of the engine. In both cases the fan speed and thus the electrical power required to operate the fans represents a parasitic load or power which must be provided in addition to the demand from the complex being served. Thus, on a day with a high wet bulb temperature, when fans would be operated on fast speed, more power would be required from either plant to provide the same power to the complex than on a day with lower wet bulb temperatures, when the fans could be operated on slow. This represents a larger fuel consumption to supply the same demand or a net plant efficiency decrease due to the increased fan driving power.

The operational performance of the cooling towers

selected for waste heat dissipation in the extraction steam turbine and dual-fuel engine plant configurations will have a significant impact on the feasibility analysis only if the annual fuel cost difference between the competing plants, without variation in the waste heat dissipation system parameters, is less than the potential fuel consumption or savings due to the possible variations. An assessment of the magnitude of the effect of cooling system performance on plant efficiency is important, since it will indicate whether the simulation should incorporate a cooling tower performance model. If, for the feasibility analysis, comparative fuel cost margins -- from a simulation with yearly average cooling system conditions -- are larger than conceivable variations from cooling system performance, then the additional expense of simulating cooling tower operation is unncessary. Calculations detailed in Appendix H show that the maximum fuel consumption variation due to operation of the cooling tower fans (fast speed fans compared to no fan operation) is 0.55% and 0.75% for the 10 MW(e) and 15 MW(e) extraction steam turbine plants respectively, and 0.67% for the dual-fuel engine installation. These values based on operation at the 1976 MIT average load conditions indicate that changes in the parasitic load due to the cooling tower fans can be neglected in the feasibility analysis unless comparative fuel costs on the order or less than 1% are important.

The effect of the cooling water temperature variations on the performance of the extraction steam turbine plants is

currently impossible to determine accurately due to a lack of information from the vendor on the turbines' performance at other than the rated 3 inch Hg condenser pressure. This off-design performance is highly dependent upon the specific turbine design and is difficult to postulate. However, considering that the cooling system automatic control will maintain cooling water temperature within a band of several degrees by regulating fan speed and bypassing the tower at extremely low temperatures, the condensing temperature should also remain within a comparable band. With a control system which maintains the temperature of the cooling water to the condenser within a conservatively large 10 F° band (the present system for the cooling towers at MIT maintains a 6 F° range), condensing temperatures should be no more than 10 F° less than the design maximum of 115°F, which corresponds to the 3 inch Hg rated condenser pressure. The lowest possible condensing temperature would then be 105°F, corresponding to 2.242 inches Hg condenser pressure. This condenser pressure difference of 0.758 inch Hg represents an isentropic exhaust enthalpy difference of only about 16 BTU per pound of exhaust steam. The enthalpy drop across just the low pressure stage of either extraction steam turbine is on the order of 300 BTU per pound of steam, so that the change in turbine power due to the variation of cooling water temperature would be on the order of .5%. At the 1976 average electrical demand of 9.567 MW(e) the .5% increase in electrical power production from the extraction steam turbine plant corresponds to about 1.5% less

fuel consumption. The calculations for this approximation are also included in Appendix H. This 1.5% fuel savings would be fully realized only in the extremely unrealistic condition where the ambient wet bulb temperature was so low that the cooling system was operating about its low temperature regularly, by-passing the cooling tower during the entire year!

Thus, with the extraction steam turbine configurations, the extreme maximum variation in fuel consumption due to cooling system performance, including parasitic power and condenser back pressure, is on the order of 2%. Unless this variation would change the comparative evaluation of the total energy plants there is no need to incorporate a cooling tower performance model in the simulation.

The inclusion of a cooling tower performance model in the simulation routine is highly unlikely as the initial simulation (25) has shown fuel cost for the extraction steam turbine plant to exceed the same cost for the competing plant configurations by more than 19%. In the unlikely event that later developments require the simulation of cooling tower off-design performance, the model used would have to be very accurate since small magnitude effects would obviously be important. For this case it is recommended that either the model developed by Guyer and Golay (66) be used or that very specific data be obtained from the vendor for a particular tower at the MIT conditions. Both of these options are expected to be expensive either in computer time for the case

of the model or in design and testing charges for the specific vendor data; however, either of these alternatives would be preferable to an approximate model such as could be developed from the general cooling tower performance characteristics available from the Cooling Tower Institute (67) or others (68).

CHAPTER IV. CONCLUSIONS AND RECOMMENDATIONS

It has been concluded that the application of thermal energy storage is not a realistic concept for inclusion in a total energy system design for MIT and that the waste heat dissipation systems required for the competing plant configurations will have a minimal impact on both the capital and operating cost comparisons in the feasibility study.

Thermal energy storage, though an appealing concept, is not attractive for the proposed plants at MIT due to the necessity of continuing to use steam as the thermal power distribution medium and due to the frequency with which the thermal power generation excess occurs.

There is available space on the MIT site for construction of adequate cooling facilities with any of the plant configurations. The projected cooling system initial cost is about 9.5% of the capital cost of the facility, projected by Was (25) to be $12,549,000 for the 15 MW(e) extraction steam installation which is the configuration in which the cooling system has the largest impact.

The cooling water system for the dual fuel engine configuration represents only 0.96% of the total projected capital cost of $5,530,000. Since the recommended cooling system for the gas turbine plant relies on the heating of makeup feedwater its cost is properly in the plant cost and is negligible. The operating cost of the cooling systems is not considered to be significant since the maximum operating

power requirement, including power for the cooling water pump and fans operating on fast speed, is only 350 KW for the 15 MW(e) extraction steam turbine and 90 KW for the 3 unit dual-fuel engine configuration, where only fan power is considered since the water pump is driven directly by the engine. This parasitic power is 3.6% and 0.9% of the 1976 MIT average electrical demand and, since the preliminary study indicated that the annual fuel cost associated with the extraction steam turbine plant was 19% greater than that of the dual-fuel engine plant, detailed consideration of the parasitic power cost will only make the steam plant more unattractive.

Considering the magnitudes of the maximum possible effects of cooling system performance on plant operation, it is recommended that the simulation of plant operation simply assume a waste heat dissipation capability which would lead to plant parameters compatible with design performance and that parasitic power be included where applicable as a constant value corresponding to continuous pump operation and one-half the value of continuous fast speed fan operation. Unless the extraction steam turbine plant costs differ from those of the competing plants by less than 2% (allowing for a linear combination of 1.0% maximum change in fuel consumption due to condenser temperature variations and 0.55% due to fan speed variation), there is no need to consider cooling system performance in a more elaborate manner. The only other design

affected by cooling system performance is the dual-fuel engine plant where only the variation of fan operating speed is at all important with a maximum change in fuel consumption of 0.67%. If fuel consumption differences on this order of magnitude became important a very detailed design would be required since few of the estimates used in the entire study are considered to be this precise.

The matching of the approximate performance characteristics of the plant configurations for each of the three prime movers with the design 1976 load profiles and an evaluation of the waste heat dissipation options has led to the following conceptual system designs:

1. The rated 10 MW(e) extraction steam turbine plant controlled to match electrical and thermal power generation to the demand when turbine limitations allow. When the minimum extraction steam flow limit is encountered the plant control would be shifted to providing the thermal power required. With the maximum allowed electrical generation less than the demand during this period a portion of the power to supply the electrical load would have to be purchased from the commercial grid. This system includes an induced-draft counterflow cooling tower. The design and cost data for the waste heat dissipation system are listed in Table 4.1.

2. Two of the rated 10 MW(e) extraction steam turbine units with no tie to the commercial grid. Each unit

TABLE 4.1

PARAMETERS OF THE COOLING SYSTEM FOR EMPLOYMENT WITH A 10 MW(e) EXTRACTION STEAM TURBINE

Design Parameters:	
Approach	9 F° @ 76°F design wet bulb temperature
Range	25 F°
Flow Rate	7726 gpm
Cooling Tower:	
Basin Area	1514 sq. ft.
Tower Units	10739
Fan Power	172 h.p. @ fast speed (2-speed fan motor)
Cooling Water Pump:	146 hp continuous activity
Condenser:	
Heat Transfer Area	10,648 sq. ft.
Water Velocity in Tubes	6 fps
Log Mean Temperature Difference	13.95 F°
Approximate Capital Cost - Materials and Installation:	
Cooling Tower	$ 257,736
Cold Water Basin	23,497
Condenser	63,888
Pump & Piping	550,114
TOTAL	$ 895,235

would use a separate cooling system exactly as with the single turbine installation. This configuration is not expected to be attractive.

3. The 15 MW(e) extraction steam turbine plant with a larger cooling tower similar to that used for the smaller plant. Design and cost data for this cooling system are summarized in Table 4.2.
4. A rated 15 to 20 MW(e) gas turbine plant with fully-fired waste heat recovery boilers. The plant design should include preheating of makeup feed by cooling of the lubricating system
5. Three 6.94 MW(e) dual-fuel engines with unfired waste heat recovery boilers and 60 MW(t), (173,110 lb/hr steam flow at 200 psig at 420°F), of conventional boiler capacity from retention of a portion of the present facility. Each engine would be cooled from a separate prefabricated cooling tower. Design and cost information for this cooling tower are presented in Table 4.3.

It is expected that the study of these configurations providing for the 1976 load will indicate the most attractive design to pursue in simulations with an expanded demand to account for the growth of the institute.

TABLE 4.2

PARAMETERS OF THE COOLING SYSTEM FOR EMPLOYMENT WITH A 15 MW(e) EXTRACTION STEAM TURBINE

Design Parameters:	
Approach	9 F° @ 76°F design wet bulb temperature
Range	25 F°
Flow Rate	11,364 gpm
Cooling Tower:	
Basin Area	2227 sq. ft.
Tower Units	15,796
Fan Power	253 hp @ fast speed (2-speed fan motor)
Cooling Water Pump:	215 hp continuous duty
Condenser:	
Heat Transfer Area	15,662 sq. ft.
Water Velocity in Tubes	6 fps
Log Mean Temperature Difference	13.95 F°
Approximate Capital Cost:	
Cooling Tower	$ 379,104
Cold Water Basin	34,563
Condenser	93,972
Pump & Piping	678,718
TOTAL	$1,186,356

TABLE 4.3

PARAMETERS OF COOLING TOWER FOR EMPLOYMENT WITH A DUAL-FUEL ENGINE

Design Parameters:	
Approach	9 F° @ 76°F design wet bulb temperature
Range	27 F°
Flow Rate	1200 gpm
Cooling Tower:	
Basin Area:	130.55 sq. ft. (9.44' x 13.83')
Height	17.44 ft.
Shipping Weight	17,500 lbs.
Fan Power:	40 hp @ fast speed (2-speed fan motor)
Capital Cost including Shipping and Installation:	
One Tower	$18,000

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APPENDIX A.

1976 MIT Power Demand Profiles for Extreme Days and Monthly Average

The data tabulated below were compiled from the operating logs of the MIT Central Heating and Cooling Facility with the cooperation of Mr. George Reid, Plant Supervisor.

TABLE A.1

HOURLY POWER DEMAND FOR FRIDAY, AUGUST 31, 1976
(Day during which maximum electric power demand occurred)

Hour	Electrical Demand MW(e)	Thermal Demand MW(t)
0100	8.76	21.48
0200	8.58	20.83
0300	8.46	20.51
0400	8.37	20.18
0500	8.34	19.86
0600	8.16	20.18
0700	8.40	21.48
0800	9.96	22.79
0900	12.21	27.02
1000	13.68	32.55
1100	15.24 - 1976 maximum	30.27
1200 (noon)	14.20	30.59
1300	14.24	30.59
1400	14.24	30.59
1500	14.24	29.94
1600	14.00	28.97
1700	12.99	29.62
1800	12.06	28.32
1900	10.92	27.02
2000	10.74	25.39
2100	10.38	24.41
2200	10.25	23.44
2300	9.82	22.46
2400	9.06	21.81

TABLE A.2

HOURLY POWER DEMAND FOR FRIDAY, JANUARY 23, 1976
(Day during which maximum thermal power demand occurred)

Hour	Electrical Demand MW(e)		Thermal Demand MW(t)	
0100	8.04		62.11	
0200	7.68		62.45	
0300	7.44		63.13	
0400	7.32		63.13	
0500	7.32		62.45	
0600	7.14		63.45	
0700	7.08		63.47	
0800	7.98		66.88	
0900	9.96	- 1976 maximum difference -	69.62	
1000	11.46		68.93	
1100	12.18		69.96	- 1976 maximum
1200 (noon)	12.45		69.62	
1300	12.60		69.62	
1400	12.48		66.88	
1500	12.48		65.52	
1600	12.54		66.88	
1700	12.48		66.88	
1800	11.52		65.86	
1900	10.80		66.20	
2000	10.38		65.52	
2100	9.84		65.18	
2200	9.66		64.50	
2300	9.18		64.50	
2400	8.76		64.50	

TABLE A.3

HOURLY POWER DEMAND FOR SATURDAY, MAY 22, 1976
(Day during which minimum electric power demand occurred)

Hour	Electrical Demand MW(e)	Thermal Demand MW(t)
0100	7.44	16.60
0200	7.23	16.93
0300	7.14	17.58
0400	6.96	17.58
0500	6.84	17.90
0600	5.94	17.91
0700	5.52 - 1976 minimum	17.91
0800	5.64	18.29
0900	6.90	18.55
1000	7.26	18.88
1100	7.50	18.56
1200	7.74	17.90
1300	8.34	15.58
1400	8.28	16.60
1500	8.43	15.95
1600	8.28	15.63
1700	8.34	17.25
1800	7.92	18.23
1900	8.04	17.90
2000	8.04	16.93
2100	8.10	16.93
2200	8.04	16.92
2300	7.80	16.60
2400	7.50	16.60

TABLE A.4

HOURLY POWER DEMAND FOR SUNDAY, SEPTEMBER 12, 1976
(Day during which minimum thermal power demand occurred)

Hour	Electrical Demand MW(e)		Thermal Demand MW(t)	
0100	7.74		13.20	
0200	7.56		13.87	
0300	7.44		13.86	
0400	7.26		13.20	
0500	7.20		13.20	
0600	7.14		13.20	
0700	7.02		13.53	
0800	6.96		16.53	
0900	7.56		16.50	
1000	7.68		14.85	
1100	8.04		14.85	
1200	8.28		13.86	
1300	8.58		14.19	
1400	8.64		14.52	
1500	8.58		13.86	
1600	8.70		14.85	
1700	8.58		15.51	
1800	8.40		15.51	
1900	8.40		15.18	
2000	8.70		15.18	
2100	8.70		13.86	
2200	8.58 –	1976 minimum difference –	12.54	– 1976 minimum
2300	8.28		12.54	
2400	7.86		13.20	

TABLE A.5

MONTHLY AVERAGE POWER DEMAND AND TEMPERATURE FOR 1976

Month	Avg. Electrical Demand, MW(e)	Avg. Thermal Demand, MW(t)	°F Avg. Temp.
Jan.	9.08	48.30	26.1
Feb.	9.31	40.88	37.3
Mar.	9.15	36.09	41.2
April	9.38	26.15	55.1
May	9.09	20.29	60.2
June	10.16	20.83	73.4
July	10.27	20.56	72.9
Aug.	10.08	20.05	72.0
Sept.	9.83	17.32	64.9
Oct.	9.82	28.32	52.3
Nov.	9.45	36.72	41.9
Dec.	9.11	47.03	29.0

APPENDIX B.

MIT Load Growth Projection

The following explanation of the basis on which the projection of MIT's power demand was made is taken directly from Was (25):

> "The forecast of campus development to the year 2000 was provided by R. Thompson of the MIT Planning Office (Table A.1) and was used as a basis for future load growth prediction, done by Mathewson (44). Once building construction projections are made, steam and electrical loads associated with these buildings must be estimated. This was not a straightforward task since buildings now on campus have electrical loads that vary from 10.5 KWH/sf. to 54.0 KWH/sf. and steam loads that vary from 97 lbm/sf. to 303 lbm/sf., the higher figures reflecting the new, more energy-wasteful buildings.
>
> To solve this problem, buildings on campus were classified into six types:
>
> 1. classroom/faculty office
> 2. classroom/labs/workshops
> 3. administrative offices
> 4. living (dorm/apartment)
> 5. athletic
> 6. computer/electronics
>
> Data on steam and electrical consumption in these buildings was collected for the years 1972-73 and

1975-76. As a rule, the lower consumption figures for steam and electricity were chosen for load growth studies to accommodate the Institute's plans to build more energy conservative buildings in the future. These loads were tabulated for each building type at five-year intervals from 1980 to 2000 and appear in Table B.2, electrical loads; and Table B.3, thermal loads. Included in these tables are calculations accounting for a 12% reduction in load due to Facilities Management."

TABLE B.1

MIT PROJECTED BUILDING CONSTRUCTION

PARCEL IDENTIFICATION	GND AREA IN SQ.FT.	RECOMMENDED BLDG. AREA IN SQ. FT.	SQ. FT. DEVELOPMENT IN 5 YEAR INCREMENTS BY SPACE TYPE			COMMENTS
30, 36–44 Mem. Drive	71,000	175,000	100,000	1995	General purpose academic - classrooms/offices.	1990 earliest availability of land.
			75,000	2000	Residential	
70 Mem. Drive	60,000	180,000	180,000	1990	Residential	Existing bldg. will probably be renovated by 1980. For interim period, 55,000 sf.
East Campus Area	600,000	1,400,000	115,000	1980	Renovate E40 office.	250,000 sf. existing-E18, E19.
			135,000	1985	Renovate Kendall bldg. - office.	
			200,000	1980	New lab/office	
			150,000	1985	General purpose academic.	
			150,000	1990	" " "	
			100,000	1995	" " "	
			100,000	2000	" " "	
			85,000	1985	Special purpose - ans.	
			35,000	1990	" " "	
			60,000	1995	" " "	
			30,000	2000	" " "	

TABLE B.1 cont'd

PARCEL IDENTIFICATION	GND AREA IN SQ. FT.	RECOMMENDED BLDG. AREA IN SQ. FT.	SQ. FT. DEVELOPMENT IN 5 YEAR INCREMENTS BY SPACE TYPE	COMMENTS
Carr/East	286,000	500,000	200,000 1985 - Laboratory academic. 300,000 1995 - General purpose academic.	Replacement of bldg. 20 @ 200,000 sf. plus additional new space.
Bldg. 12 site	100,000	100,000	100,000 1985 - General purpose academic.	Proposed to remove ground floor of bldg. 24 (west wing) @ 18,000 sf. plus bldg. 31 C 62,000 sf. as part of this development.
Northesast sector from Mass. Ave. to Main St. and away to Vassar St.	300,000	600,000	100,000 1990 - Physical plant service garages, power plant, etc. 100,000 1995 - General purpose academic. 300,000 Parking structures.	Existing to remain: 41, 42, 42A, 42B, 43, N16, 48 for 125,000 sf. One third existing.
158 - 168 Mass. Ave.	22,500	100,000	100,000 1995 - Office/retail.	
Bexley/Cays	33,600	75,000	75,000 1990 - Special Purpose academic. Facility - arts?	

TABLE B.1 cont'd

PARCEL IDENTIFICATION	GND AREA IN SQ. FT.	RECOMMENDED BLDG. AREA IN SQ. FT.	SQ. FT. DEVELOPMENT IN 5 YEAR INCREMENTS BY SPACE TYPE	COMMENTS
Athletics		225,000	100,000 1985 - Ice rink, field house. 125,000 1985 - Gym, pool, lockers.	
180-190 Albany St.	55,000	75,000	75,000 1980 - Magnet lab expansion.	Renovate existing bldgs.
Vassar St., West Campus	214,500	460,000	175,000 1990 - Dormitory 115,000 1995 - " 115,000 1995 - " 115,000 2000 - "	
New House, West Campus	56,000	120,000	120,000 1985 - Dormitory	
		4,000,000	1,335,000 sf. Pre-1985 total	

TABLE B.2

PROJECTED ELECTRICAL LOAD GROWTH (Loads in KW)

	1976	1980	1985	1990	1995	2000
Bldg. Type 1			314	188	753	126
Type 2		731	683	264	384	
Type 3			108			
Type 4			110	270		
Type 5			115			
Subtotal	--	731	1330	722	1347	299
Running Total	--	731	2061	2783	4130	4429
Average Load	9567	10298	11628	12350	13697	13996
Peak Load	15240	16404	18523	19674	21819	22296
Load Increase, %	--	7.64%	21.54%	29.09%	43.17%	46.29%
With 12% Load Reduction:						
Average Load	8419	9062	10233	10868	12053	12317
Peak Load	13411	14436	16301	17313	19201	19620
Load Increase, %	-12.0%	-5.28%	6.96%	13.60%	25.99%	28.74%

(see page 155 for key to building types)

TABLE B.3

PROJECTED THERMAL LOAD GROWTH (Loads in lbm/hr, @ 200 psi)

	1976	1980	1985	1990	1995	2000
Bldg. Type 1		2124	1741	1045	4178	696
Type 2			1985	766	1114	
Type 3			940			
Type 4			890	2189	1707	1410
Type 5			1387			
Subtotal	--	2124	6943	4000	6999	2106
Running Total	--	2124	9067	13067	20066	22172
Average Load	95294	97418	104361	108361	115360	117466
Peak Load	217000	221821	237630	246738	262675	267470
Load Increase, %	--	2.23%	9.51%	13.71%	21.06%	23.27%
With 12% Load Reduction:						
Average Load	83859	85728	91838	95358	101517	103370
Peak Load	190946	195203	209115	217130	231154	235374
Load Increase, %	-12.00%	-10.04%	-3.63%	0.07%	6.53%	8.47%

(see page 155 for key to building types)

APPENDIX C.

Calculations for Extraction Steam Turbine Performance

Calculation of conversion factor for extraction steam flow to thermal power:

With the extraction steam at 200 psig with 160 F^{o} superheat,

extraction steam enthalpy = 1293.2 BTU/lbm

(the state of the extraction steam actually varies slightly over the range of turbine load conditions with 200 psig 160 F^{o} superheat as a minimum design condition)

With condensate return from the thermal load at an average temperature of 160°F at 60 psig and 8% makeup at 50°F, the condensate enthalpy is about 110 BTU/lbm. By considering the makeup feed enthalpy in this manner a minimum condensate enthalpy is determined which will yield a maximum thermal power to the load, with the following calculations:

Enthalpy change of the extraction steam supplied to the load = Δh

$$\Delta h = 1293.2 - 110 \text{ BTU/lbm}$$
$$\Delta h = .34667 \text{ kw-hr/lbm}$$

With this conversion the following equivalencies are determined:

extraction flow	thermal supply
50,000 lb/hr	17.33 MW(t)
100,000 lb/hr	34.67 MW(t)
150,000 lb/hr	52.00 MW(t)
200,000 lb/hr	69.33 MW(t)

TABLE C.1

SUPPLY/DEMAND MISMATCH FOR 10 MW(e) EXTRACTION STEAM TURBINE PLANT ON DAY OF 1976 MAXIMUM THERMAL DEMAND (January 23, 1976)

Hr.	Demand		Allowed Supply/Mismatch	
	Electrical	Thermal	*Electrical	†Thermal
01	8.76	21.48		
	8.58	20.83		
	8.46	20.51		
	8.37	20.18	(where blank -- demand, supply match)	
	8.34	19.86		
06	8.16	20.18		
	8.40	21.48		
	9.96	22.79		
	12.21	27.02		
	13.68	32.55	12.95/-.73	39.37/7.32
	15.24	30.27	12.78/-2.46	57.89/27.62
12	14.20	30.60	12.80/-1.40	45.07/14.47
	14.24	30.60	12.80/-1.44	45.53/14.93
	14.24	30.60	12.80/-1.44	45.53/14.93
	14.24	29.95	12.76/-1.48	45.53/15.58
	14.00	28.97	12.65/-1.35	42.99/14.02
	12.99	29.62	12.70/-.029	32.24/2.62
18	12.06	28.32		
	10.92	27.02		
	10.74	25.39		
	10.38	24.41		
	10.25	23.44		
	9.82	22.46		
24	9.06	21.81		

*Control on thermal demand.

†Control on electrical demand.

APPENDIX D.

Determination of Thermal vs. Electrical Power Generation Capability for Gas Turbine Plant With Waste Heat Recovery Boiler

An energy balance on the waste heat recovery boiler, neglecting losses to ambient, yields:

$$\dot{m}_{\text{exhaust gas}}(h_{in} - h_{out}) = \dot{m}_{steam}(h_{steam} - h_{feed}) \tag{1}$$

or

$$\dot{m}_{exh}\, h_{exh} - \dot{m}_{exh}\, h_{stack} = \dot{m}_{steam}\,\Delta h_{boiler} \tag{2}$$

Was and Mathewson (25) determined:

$$\dot{m}_{exh}\, h_{exh} =$$

$$[(64.9738) + (.005367)x + (8.7823 \times 10^{-8})x^2](\frac{BTU}{hr} \times 10^6) \tag{3}$$

$$\dot{m}_{exh} =$$

$$[(3365) + (.3809)x - (6.1048 \times 10^{-6})x^2](\frac{lbm}{min}) \tag{4}$$

where x = plant electric output in KW(e).

Allowing for 3% losses of the gas turbine rejected heat to include losses to ambient and leakage flow through the waste heat boiler's bypass damper, and approximating the enthalpy of the exhaust gas at the outlet of the waste heat boiler at 85 BTU/lbm:

$$0.97\ \dot{m}_{exh}\, h_{exh} - \dot{m}_{exh}\, h_{stack} =$$

$$[(4.5863 \times 10^7) + (3.2634 \times 10^3)x + (.1163)x^2](\frac{BTU}{hr}) \tag{5}$$

With the efficiency of the waste heat recovery boiler, η_{whb}, defined as

$$\eta_{whb} = \frac{\dot{m}_{stm}\ \Delta h_{boiler}}{(.97\,\dot{m}_{exh}\,h_{exh}) - (\dot{m}_{exh}\,h_{stack})} \tag{6}$$

$$\dot{m}_{stm} = \frac{\eta_{whb}}{\Delta h_{boiler}}[(.97\,\dot{m}_{exh}\,h_{exh}) - (\dot{m}_{exh}\,h_{stack})] \tag{7}$$

With boiler enthalpy difference for the MIT plant which has been previously determined to be 1111 BTU/lbm steam and the conversion ratio for steam flow rate to thermal power set at $.3255 \frac{\text{KW(t)-hr}}{\text{lbm stm}}$ for the MIT conditions thermal power is obtained as a function of electrical power generated by substitution of (5) into (7).

Thermal Power Available =

$$[(1.3437 \times 10^4) + (.95611)x + (3.4073 \times 10^{-5})x^2]\eta_{whb} \tag{8}$$

Typically boiler efficiencies vary as a function of steaming rate from a minimum of about .8 at 20% rated load to a maximum of about .9 at about 80% rated load with an average of about .86 for loadings greater than 50%. Using the .86 average value for η_{whb}

Thermal Power Available KW(t) =

$$(1.1556 \times 10^4) + (8.2225 \times 10^{-1})x + (2.9303 \times 10^{-5})x^2 \tag{9}$$

APPENDIX E.

Dual-Fuel Engine Plant Performance Calculations

(For design data on typical engine system see Table 2.2)

At Full Load:

Exhaust Heat =

$$(2220 \frac{\text{BTU}}{\text{BHP-hr}})(9630\ \text{BHP}) = 2.138 \times 10^7 \frac{\text{BTU}}{\text{hr}} = 6.26\ \text{MW(t)}$$

Exhaust Heat Recovered = $12850 \frac{\text{lb stm}}{\text{hr}} \times$ (Δh enthalpy in boiler)

$$\Delta h \text{ Enthalpy in Boiler} = \Delta h = h_{stm} - h_{feed}$$

$$= 1223.7 \frac{\text{BTU}}{\text{lbm}} - 110 \frac{\text{BTU}}{\text{lbm}}$$

(steam at 200 psig 420°F, feed at 92% 165°F, 8% 50°F)

$$= 1113.7 \frac{\text{BTU}}{\text{lbm}}$$

$$\text{Exhaust Heat Recovered} = (12850)(1113.7) \frac{\text{BTU}}{\text{hr}}$$

Exhaust Waste Heat Recovery Effectiveness =

$$\frac{\text{Exhaust Heat Recovered}}{\text{Exhaust Heat}} = 66.94\%$$

Output per Unit:

% Electrical Load	Electrical Power Out	Thermal Power Out
50	3.47 MW(e)	2.06 MW(t)
75	5.21 MW(e)	2.92 MW(t)
100	6.94 MW(e)	4.19 MW(t)

APPENDIX F.

Incremental Cost Basis for Cooling Towers at MIT

Towers 3 and 4 Incremental Pricing Exclusive of Basin and External Piping or Wiring:

	Tower 3	Tower 4
Ambient Wet Bulb Temperature °F	76	76
Design Range F°	32	32
Design Approach F°	9	9
Rating Factor from Fig. 3.8	1.57	1.57
Design Water Flow, gpm	7000	7950
Required Tower Units, TU	10990	12482
First Cost	\$175,777(1971)	\$234,376(1973)
Equivalent Cost 1977	\$249,343	\$295,894
Incremental Cost per TU	\$22.69/TU	\$23.71/TU

Basin Sizing and Cost - Basin Constructed in 1971 for both Towers 3 and 4:

Basin Dimensions	50' 1.5" x 66' 1.5" = 3314.52 ft^2
Incremental Basin Requirements, ft^2/TU	$\frac{3314.52}{10990 + 12482} = .141\ ft^2/TU$
Basin Cost (1971)	\$36,260
Equivalent Cost 1977	\$51,436
Cost/ft^2	\$15.52

	Tower 3	Tower 4
Fan Horsepower Requirements:		
Rated Air Flow CFM	421,590	411,724
Total Fan Horsepower Installed (at full speed)	250	200
Fan Horsepower/TU	.0227 hp/TU	.0160 hp/TU
Total Water Pumping Head at Stated Flow for Tower Alone (measured at plane of bottom of cold water basin)	44.55 ft water	42.0 ft water
Height of Bottom of Cold Water Basin Above Pump Foundation (static head)	26 ft.	26 ft.

Tower 4 - Pump & Piping Costs:

Costs Variable with Flow Rate: (shown for 7950 gpm flow)

Pump & Motor	$ 15,000	
Piping & Fittings	180,000	
Pipe Covering	15,000	
	$ 210,000	(1972)
Escalated to 1977	$ 281,000	

Fixed Costs:

Controls & Instrumentation	$ 13,200	
Excavation	96,000	
Under R.R. Track Work	21,000	
Electric Wiring & Installation	57,000	
Chemical Feed System	7,500	
Piping Installation	10,000	
	$ 206,700	(1972)
Escalated to 1977	$ 277,000	

APPENDIX G.

Cooling System Design Parameter Optimization Calculations for Extraction Steam Turbine Plant

The tabulated values were determined by performing the following sequence of calculations:

1. For each approach calculate the range

   Range = (Condensing Temperature) - (Terminal Temperature Difference) - (Design Wet Bulb Temperature) - (Approach)

   $$\text{Range} = (115^{\circ}\text{F}) - (5^{\circ}\text{F}) - (76^{\circ}\text{F}) - (\text{Approach})$$

   $$\text{Range} = 34^{\circ}\text{F} - \text{Approach}$$

2. Determine the cooling water flow rate in gpm

   $$\text{Flow Rate} = \frac{\text{Rejected Heat Rate}}{\text{Range}} \text{ x (Specific Heat of Water)}$$

   Case 1: Flow Rate =

   $$\frac{(28.29\ \text{MW}) \text{ x } (3413 \text{ x } 10^3\ \frac{\text{BTU}}{\text{mw-hr}})}{(\text{Range}) \text{ x } (1\ \frac{\text{BTU}}{\text{lbm}^{\circ}\text{F}})} \text{ x } \frac{\text{gal}}{8.331\ \text{lbm}} \text{ x } \frac{\text{hr}}{60\ \text{min}}$$

   Flow Rate = 193161.63/Range

   Case 2: Flow Rate =

   $$\frac{(41.61\ \text{MW}) \text{ x } (3413 \text{ x } 10^3\ \frac{\text{BTU}}{\text{mw-hr}})}{(\text{Range}) \text{ x } (1\ \frac{\text{BTU}}{\text{lbm}^{\circ}\text{F}})} \text{ x } \frac{\text{gal}}{8.331\ \text{lbm}} \text{ x } \frac{\text{hr}}{60\ \text{min}}$$

   Flow Rate = 284109.41/Range

3. With the chosen approach and calculated range find the cooling tower "Rating Factor", RF, on the appropriate graph of Fig. 3.8.

4. Calculate the tower units, TU:

   TU = (RF) x (Flow Rate)

5. Determine cooling tower initial cost at \$24/TU.

6. Calculate required basin area at 0.141 ft^2/TU.

7. Compute basin cost at \$15.52/ft^2.

8. Calculate the condenser log mean temperature difference

   LMTD =

$$\frac{\text{Range}}{\ln\left(\dfrac{\text{Condensing Temp.} - \text{Cooling Water Inlet to Cond. Temp.}}{\text{Condensing Temp.} - \text{Cooling Water Outlet from Cond.Temp}}\right)}$$

$$= \frac{\text{Range}}{\ln\left(\dfrac{115^\circ - 76^\circ - \text{Approach}}{115^\circ - 76^\circ - \text{Approach} - \text{Range}}\right)}$$

$$= \frac{\text{Range}}{\ln\left(\dfrac{39^\circ - \text{Approach}}{39^\circ - \text{Approach} - \text{Range}}\right)} = \frac{\text{Range}}{\ln\left(\dfrac{39^\circ - \text{Approach}}{5^\circ}\right)}$$

9. Determine the required condenser heat transfer surface area, A

$$A = \frac{\text{Rejected Heat Rate}}{U \times \text{LMTD}}$$

Case 1: $$A = \frac{28.29\ \text{MW(t)} \times (3413 \times 10^3\ \frac{\text{BTU}}{\text{mw-hr}})}{(650\ \frac{\text{BTU}}{\text{hr-ft}^2\text{-}^\circ\text{F}}) \times \text{LMTD}}$$

$$A = \frac{148544.26}{\text{LMTD}}\ \text{ft}^2$$

Case 2: $$A = \frac{41.61\ \text{MW(t)} \times (3413 \times 10^3\ \frac{\text{BTU}}{\text{mw-hr}})}{(650\ \frac{\text{BTU}}{\text{hr-ft}^2\text{-}^\circ\text{F}}) \times \text{LMTD}}$$

$$A = \frac{218484.51}{\text{LMTD}}\ \text{ft}^2$$

10. Compute condenser first cost at $6 per square foot of required heat transfer surface area.

11. Calculate the pump and piping system initial cost

$$\text{Pump \& Piping Cost} = \$277{,}000 + (\frac{281{,}000}{7950} \times \text{Flow Rate})$$

$$= \$277{,}000 + (\frac{\$35.35}{\text{gpm}} \times \text{Flow Rate})$$

12. Sum cooling tower initial cost, basin cost, condenser first cost, and pump and piping cost to obtain total capital cost for cooling system in 1977 dollars.

13. Compute the required fan horsepower at 0.016 hp/TU.

14. Estimate yearly fan operating cost

$$\text{Fan Operating Cost} = \frac{\text{power cost}}{\text{hp-hr}} \times (\text{fan hp}) \times (\text{fan utilization})$$

$$= \frac{\$.0269}{\text{hp-hr}} \times (\text{fan hp}) \times (4380\ \text{hr/yr})$$

$$= \frac{\$117.82}{\text{hp-yr}} \times (\text{fan hp})$$

15. Calculate the required pump horsepower

Pump Horsepower = (Flow Rate) x (Head)

$$= \frac{(\text{Flow Rate, gpm}) \times 8.331 \frac{\text{lbm}}{\text{gal}} \times 75 \text{ ft}}{\frac{33000 \text{ ft lb/min}}{\text{hp}}}$$

$$(.0189 \frac{\text{hp}}{\text{gpm}}) \times (\text{Flow Rate, gpm})$$

16. Estimate the yearly pump operating cost

Pump Operating Cost =

$$\frac{(\frac{\text{Power Cost}}{\text{hp-hr}}) \times (\text{Pump Horsepower}) \times (\text{Pump Utilization})}{(\text{Pump Motor Efficiency}) \times (\text{Pump Efficiency})}$$

Pump Operating Cost =

$$\frac{(\frac{\$.0269}{\text{hp-hr}}) \times (\text{Pump Horsepower}) \times (8760 \frac{\text{hr}}{\text{yr}})}{(.92) \times (.85)}$$

Pump Operating Cost = $(\frac{\$301.34}{\text{hp-yr}}) \times (\text{Pump Horsepower})$

17. Sum fan and pump operating costs to determine yearly operating costs.

18. Compute net present worth of 30 years of operating power costs

Net Present Worth of 30 Years Operation =

$$(\text{Yearly Operating Cost}) \times \frac{(1 + i)^{30} - 1}{i(1 + i)^{30}}$$

where i = yearly interest rate.

Net Present Worth of 30 Years Operation =

13.765 x (Yearly Operating Cost)

19. Sum cooling system total capital cost and net present worth of 30 years operating power cost to get net present worth of cooling system.

20. Compute cost per year of life for cooling system and its operating power by summing yearly operating cost and yearly capital recovery of cooling system initial cost

$$\text{Cost/Year} =$$

$$\left(\frac{\text{Operating Cost}}{\text{Year}}\right) + (\text{Cooling System Initial Cost}) \times \frac{i(1+i)^{30}}{(1+i)^{30}-1}$$

$$\text{Cost/Year} =$$

$$\left(\frac{\text{Operating Cost}}{\text{Year}}\right) + .7265\,(\text{Cooling System Initial Cost})$$

TABLE G.1

COOLING SYSTEMS FOR USE WITH 10 MW(e) EXTRACTION STEAM TURBINE

28.29 MW(t) - Maximum Rejected Heat Rate

$76^{o}F$ - Design Ambient Wet Bulb Temperature

Approach	8	9	10	12	14	16	18
1. Range, F^{o}	26	25	24	22	20	18	16
2. Flow Rate, gpm	7429	7726	8048	8780	9658	10731	12073
3. Rating Factor	1.57	1.39	1.24	1.00	0.81	0.66	0.54
4. Tower Units	11664	10739	9980	8780	7823	7082	6519
5. Cooling Tower Cost, $	279,936	257,736	239,520	210,720	187,752	169,968	156,456
6. Basin Area, ft^2	1645	1514	1407	1236	1103	999	919
7. Basin Cost, $	25,530	23,497	21,837	19,183	17,119	15,504	14,263
8. LMTD, ^{o}F	14.25	13.95	13.65	13.05	12.43	11.80	11.15
9. Cond. Area, ft^2	10424	10648	10882	11383	11950	12588	13322
10. Cond. Cost, $	62,544	63,888	65,292	68,298	71,700	75,528	79,932
11. Pump & Piping Cost, $	539,615	500,114	561,497	587,373	618,410	656,341	703,781
12. Cooling System Cost, $	907,625	895,235	888,146	885,574	894,981	917,341	954,432
13. Fan Horsepower	187	172	160	140	125	113	104

TABLE G.1 cont'd

Approach	8	9	10	12	14	16	18
14. Fan Operating Cost, \$/yr	22,032	20,265	18,851	16,495	14,728	13,314	12,253
15. Pump Horsehpower	141	146	152	166	183	203	229
16. Pump Operating Cost, \$/yr	42,386	44,081	45.918	50,094	55,104	61,226	68,883
17. Yearly Operating Cost \$/yr	64,418	64,346	64,769	66,589	69,832	74,540	81,136
18. Present Worth of 30 years of Operation, \$	886,714	885,723	891,545	916,598	961,237	1,026,043	1,116,837
19. Net Present Worth, 1977 \$	1,794,339	1,780,958	1,779,691	1,802,172	1,856,218	1,943,384	2,071,269
20. Net Yearly Cost, \$	130,357	129,385	129,293	130,926	134,852	141,185	150,475

TABLE G.2

COOLING SYSTEMS FOR USE WITH 15 MW(e) EXTRACTION STEAM TURBINE

41.61 MW(t) - Maximum Rejected Heat Rate

76^{o}F - Design Ambient Wet Bulb Temperature

Approach	8	9	10	12	14	16	18
1. Range, F^{o}	26	25	24	22	20	18	16
2. Flow Rate, gpm	10927	11364	11838	12914	14205	15784	17757
3. Rating Factor	1.57	1.39	1.24	1.00	0.81	0.66	0.54
4. Tower Units	17155	15796	14679	12914	11506	10417	9589
5. Cooling Tower Cost, $	411,720	379,104	352,296	309,936	276,144	250,008	230,136
6. Basin Area, ft^2	2419	2227	2070	1821	1622	1469	1352
7. Basin Cost, $	37,543	34,563	32,157	28,262	25,173	22,799	20,983
8. LMTD, ^{o}F	14.25	13.95	13.65	13.05	12.43	11.80	11.15
9. Cond. Area, ft^2	15332	15662	16006	16742	17577	18516	19595
10. Cond. Cost, $	91,992	93,972	96,036	100,452	105,462	111,096	117,570
11. Pump & Piping Cost, $	663,269	678,717	695,473	733,510	779,147	834,964	904,710
12. Cooling System Cost, $	1,204,524	1,186,356	1,175,962	1,172,160	1,185,926	1,218,867	1,273,399
13. Fan Horsepower	274	253	235	207	184	167	153

TABLE G.2 cont'd

Approach	8	9	10	12	14	16	18
14. Fan Operating Cost, $/yr	32,283	29,808	27,688	24,389	21,679	19,676	18,026
15. Pump Horsepower	207	215	224	245	269	299	336
16. Pump Operating Cost, $/yr	62,344	64,837	67,542	73,681	81,047	90,056	101,313
17. Yearly Operating Cost $/yr	94,627	94,645	95,230	98,070	102,726	109,732	119,339
18. Present Worth of 30 years of Operation, $	1,302,541	1,302,788	1,310,841	1,349,934	1,414,023	1,510,461	1,642,701
19. Net Present Worth, 1977 $	2,507,065	2,489,144	2,486,808	2,522,094	2,599,949	2,729,328	2,916,180
20. Net Yearly Cost, $	182,136	180,834	180,664	183,227	188,884	198,283	211,851

APPENDIX H.

Calculations to Approximate the Fuel Consumption Due to Cooling System Operation and Performance

Considering the average MIT 1976 power demand, based on hourly data, of:

Average electrical load - 9.567 MW(e) - Q_e

Average thermal load - 33.029 MW(t) - Q_t

the 10 MW(e) extraction steam turbine at this load condition would require 160,000 lbm/hr of steam at 800 psig and 825°F. The enthalpy, h_o, of the steam at this condition is 1412.7 BTU/lbm. The feedwater enthalpy, h_i, would be about 110 BTU/lbm corresponding to 165°F condensate return temperature with 8% makeup feedwater at 80°F. With an approximate boiler efficiency of 86% the energy, Q_{in}, supplied to the boiler is

$$Q_{in} = \frac{160{,}000 \frac{\text{lbm}}{\text{hr}} (1412.7 - 110) \frac{\text{BTU}}{\text{lbm}}}{.86} \times \frac{\text{MW}}{3.413 \times 10^6 \frac{\text{BTU}}{\text{hr}}}$$

$$Q_{in} = 71.012 \text{ MW}$$

The net plant efficiency is, then

$$\eta_{net} = \frac{Q_e + Q_t}{Q_{in}} = \frac{9.567 + 33.029}{71.012} = 59.99\%$$

With electrical generation efficiency, η_e, of

$$\eta_e = \frac{Q_e}{Q_{in} - Q_t} = \frac{9.567}{71.012 - 33.029} = 25.188\%$$

The above calculations made no allowance for cooling tower fan power. Assuming 200 hp required for fan operation on fast speed (actual design value is 172 hp) and the recommended (45) motor drive efficiency of 92%, the parasitic electrical power, Q_{ef}, required for fan operation at fast speed is

$$Q_{ef} = \frac{200 \text{ hp} \times .7457 \frac{\text{KW}}{\text{hp}}}{.25} = .162 \text{ MW(e)}$$

At the approximate 25% electrical generation efficiency this parasitic power requires an added energy input to the boiler Q' of

$$Q' = \frac{.162 \text{ MW(e)}}{.25} = .648 \text{ MW}$$

yielding a net plant efficiency, $\eta_{net,f}$, with cooling tower fans operating on fast speed of

$$\eta_{net,f} = \frac{Q_e + Q_t}{Q_{in} + Q'} = \frac{9.567 + 33.029}{71.012 + .648} = 59.44\%$$

Thus a maximum difference of 0.55% fuel consumption is possible for the 10 MW(e) extraction steam turbine supplying the average 1976 demand over the widest possible variation of cooling tower fan speed, off to fast.

The same sequence of calculations for the 15 MW(e) extraction steam turbine plant yields:

Boiler Steam Flow - 162,000 lbm/hr

$$Q_{in} = 71.899 \text{ MW}$$

$$\eta_{net} = \frac{9.567 + 33.029}{71.899} = 59.24\%$$

$$\eta e = \frac{9.567}{71.899 - 33.029} = 24.61\%$$

with fast speed fan horsepower of 275 (actual design 753)

$$Q_{e,f} = \frac{275 \times .7457}{.92} = .223 \text{ MW(e)}$$

$$Q' = \frac{.223}{.24} = .929 \text{ MW}$$

$$\eta_{net,f} = \frac{9.567 + 33.029}{71.899 + .929} = 58.49\%$$

Fuel consumption difference for 15 MW(e) plant - .75%.

Since the dual-fuel engine plant utilizes waste heat recovery to generate thermal power, the electrical generation efficiency may be considered alone. For small changes in electric power generated the generation efficiency will be approximately constant so that the fractional increase in fuel consumption due to an increase in parasitic power will be the same as the fractional increase in required power. With 40 horsepower cooling tower fan the maximum parasitic power change due to fan speed is

$$\Delta Q = \frac{40 \text{ hp} \times .7457 \frac{\text{kw}}{\text{hp}}}{.92} = .032 \text{ MW/fan}$$

which includes a 92% efficient motor and fan drive.

With no fan operating, $Q_e = 9.567$ MW, at the average demand, and with 2 fans on fast speed $Q_{e,f} = 9.631$ MW or

$$\frac{Q_{e,f}}{Q_e} = 0.67\%$$

which represents the maximum fuel consumption variation for the dual-fuel engine plant operating to supply the average 1976 MIT demand.

With a possible .5% increase in electrical power production with constant boiler firing rate at average MIT demand conditions due to changing condenser temperature the net plant efficiency would increase to

$$\eta_{net} = \frac{(9.567)(1.005) + 33.029}{71.012} = 60.05\%$$

or an approximate fuel consumption change of

$$\frac{60.05\% - 59.99\%}{59.99\%} = 1.00\%$$

Thesis
P14435 Palmer
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An assessment of thermal energy storage and waste heat dissipation with total energy systems for MIT.